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HYDRAULIC POWER

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BY

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TO

SIR WILLIAM GEORGE ARMSTRONG,

C.B., F.R.S., D.C.L., LL.D.,

The Originator of the Modern Hydraulic System,

THIS WORK IS CORDIALLY INSCRIBED

IN GRATEFUL ACKNOWLEDGMENT

Of much kindness shown in bygone days

TO HIS FORMER ASSISTANT,

THE AUTHOR.

PREFACE.

THE increasing interest taken in water-pressure machinery, and the extended field which has opened out of late years for its employment, have led me to record, in a form convenient for reference, existing experience in this branch of engineering. In this task I have availed myself of the information published in the *Proceedings* of the Institution of Civil Engineers, the Institution of Mechanical Engineers, the Iron and Steel Institute, and of other Societies. I have thus not confined myself within the range of my own professional practice, but have utilised the experience of others wherever I have found that it would increase the usefulness of the book.

It affords me much pleasure to acknowledge the ready response that has invariably followed any request for particulars or for drawings to enable me to illustrate the varieties of hydraulic apparatus to which I desired to refer, and I believe that I have recognised in the proper places throughout the work my obligations to all who have thus kindly assisted me.

At the commencement, I refer briefly to the "Flow of water under pressure," and show the practical value of some interesting experiments which have recently been made, and which have enabled new formulæ to be deduced for the discharge from pipes. The employment of water-pressure mains, to transmit power through the streets of a town on the principle which I have termed "Power co-operation," is steadily gaining ground. The first works of the kind were those which I carried out in

Hull in 1876, and the promotion of similar undertakings in other towns will afford an increased field for utilising hydraulic power.

Whilst describing the most interesting types of Hydraulic Machinery, I have abstained alike from criticisms on the details of construction, and from any attempts to lay down fixed rules for the employment of any particular appliance. The conditions which render one form of appliance more suitable than another vary in almost every case, so that each requires to be dealt with according to the practical circumstances which govern it.

As my earliest experiences in this branch of my practice were gained nearly thirty years ago, when with Sir William Armstrong, I have dedicated this book to him; and his friendly acceptance of this dedication has enabled me, in these later days, to refer to an association which I look back upon with pleasure and pride.

HENRY ROBINSON.

7 Westminster Chambers, London, S.W.
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THE FLOW OF WATER UNDER PRESSURE.

THE weight of distilled water of maximum density is 62·499 lbs. at 39° Fahrenheit. The density diminishes towards the freezing point at which (in the form of ice) a cubic foot only weighs 58 lbs. Ordinary fresh water weighs 62½ lbs. per cubic foot, and sea water 64 lbs. Although water has been proved by experiment to be compressible (but very slightly) under great pressures, for all practical purposes it is taken as incompressible. The compressibility of water is one twenty-thousandth for an increase in pressure equivalent to an atmosphere.

The conditions which govern the flow of water through pipes, and into or out of closed vessels, have been the subject of observation and experiments dating from Torricelli's time to the present, but only a brief reference to them can be attempted. The investigations of Torricelli, Venturi, Bernouilli, Eytelwein, Darcy, Weisbach, Du Buât, Prony, Bazin, and more recently of Rankine, Downing, Reynolds, Unwin, Cotterill, Hamilton-Smith and others are published, and are available for those who wish to pursue the subject in detail.

Torricelli discovered in 1643 that the velocity of a fluid flowing through an orifice in a vessel, is that which a solid body

would acquire in falling a height corresponding to the distance between the level of the fluid in that vessel and the centre of the orifice. Employing symbols to represent this:—If v is taken for the velocity, H for the difference of level or head, and g for the measure of the force of gravity, or the number of feet per second at which a falling body is moving at the end of the first second: then $v = \sqrt{2gH}$, or the velocity varies as the square root of the head. If to the natural head is added an artificial pressure which can be regarded as a further head, H', the formula would be $v = \sqrt{2g(H + H')}$, provided the head is maintained by an influx of water equivalent to that which issues at the orifice. The expression $\sqrt{2gH}$ (representing the theoretical velocity) requires a deduction to be made from it to determine the actual velocity.

The quantity discharged from an orifice in a unit of time (as a second) varies with the velocity (v), and with the area of the orifice A. Therefore the theoretical discharge is found by the expression $D = A \sqrt{2gH}$. The actual discharge, however, is less than this, as the sectional area of the stream of water at the point of issue is not the exact area of the orifice itself. If the orifice be at the bottom of a vessel in which the head is uniformly maintained, it has been ascertained that the sectional area of the jet is $\frac{16}{25}$ths of the sectional area of the hole. This reduction in the sectional area of the stream of water is known as the "Vena Contracta," and it arises from various causes. It is allowed for in practice by a coefficient which has been arrived at by experiments upon orifices of different shapes. These give the mean reduction in diameter as 1 to ·80, and in sectional area as 1 to ·62. If the orifice be in a perfectly plane plate, the coefficient ranges between ·60 and ·64, seldom falling below the former or rising above the latter. The actual discharge will therefore be determined by the expression $D = 0{\cdot}62A \sqrt{2gH}$.

The following table gives the results of the experiments of various observers in the past:—

Experimenter.	Head.	Diameter of Orifice.	Coefficient.
	Feet.	Inch.	
Eytelwein	2·4	1·0	0·618
Bossut	0·6	1·0	0·649
Castel	2·7	1·2	0·629
Venturi	2·9	1·6	0·622
Rennie	1·0	1·0	0·633
"	2·0	1·0	0·619
Weisbach	2·0	1·2	0·614
"	2·0	1·6	0·607
"	0·8	1·2	0·622
"	0·8	1·6	0·614

Experiments have been recently made by Mr. James Simpson and Mr. John G. Mair, and the results (communicated to the Institution of Civil Engineers) are given in the following table:—

COEFFICIENTS OF DISCHARGE FROM CIRCULAR ORIFICES.

Temperature 51° to 55° Fahr.

Head. Inches.	Approximate Diameter of Orifice in Inches.								
	1	1¼	1½	1¾	2	2¼	2½	2¾	3
	Absolute Area in Square Feet.								
	0·00546	0·00852	0·012281	0·016749	0·021806	0·027576	0·033898	0·040933	0·049139
	Coefficients.								
9	0·616	0·614	0·616	0·610	0·616	0·612	0·607	0·607	0·609
12	0.613	0·612	C·612	0·611	0·612	0·611	0·604	0·608	0·609
15	0·613	0·614	0·610	0·608	0·612	0·608	0·605	0·605	0·606
18	0·610	0·612	0·611	0·606	0·610	0·607	0·603	0·607	0·605
21	0·612	0·611	0·611	0·605	0·611	0·605	0·604	0·607	0·605
24	0·609	0·613	0·609	0·606	0·609	0·606	0·604	0·604	0·605

These coefficients are lower than those determined by previous observers, and this is attributed to slight differences in the edges of the orifices.

Professor Unwin has deduced the following formula for the coefficient of discharge:—

$$C = 0{\cdot}6075 + \frac{0{\cdot}0098}{\sqrt{h}} - 0{\cdot}0037d,$$

where h is the head in feet, and d is the diameter in inches.

The temperature of water has a slight effect upon the coefficient of discharge from an orifice. Experiments made by Mr. Mair show that with a sharp-edged orifice $2\frac{1}{4}$ inches in diameter, and with a head of 21 inches, the coefficient was ·604 for temperatures varying from 57° up to 110°, and increased to ·607 for temperatures up to 179°. Further experiments were made with a conoidal orifice, in which there is no contraction of the jet, and in which the coefficient is one of velocity only. Professor Unwin's formula for the coefficient of resistance C_r in terms of the coefficient of velocity C_v is

$$(a)\ C_r = \frac{1}{C_v^2} - 1.$$

The results obtained in the experiments on temperature with a conoidal orifice (calculated by the method of least squares) gave the following formula for the coefficient of resistance:—

$$(b)\ C_r = 0{\cdot}0799 + 0{\cdot}000184t - 0{\cdot}0000025t^2.$$

Whence the following table is calculated:—

CONOIDAL ORIFICE ($1\frac{1}{4}$ inch in diameter).

Head.	Temperature. Fahr.	Coefficient of Velocity C_v.	Coefficient of Resistance by Formula (a).	Coefficient of Resistance by Formula (b).
Inches.	Degrees.			
21	55·0	0·961	0·0828	0·0825
,,	99·5	0·965	0·0739	0·0736
,,	110·1	0·967	0·0694	0·0698
,,	119·0	0·9685	0·0661	0·0664
,,	170·0	0·981	0·0391	0·0390

Where the orifice is in a pipe, and the water is issuing to the air, the coefficient depends on the proportion that the area of the pipe bears to that of the orifice. Professor Rankine investigated this, and deduced the following formula:—

$$C = \frac{1}{\sqrt{2{\cdot}618 - 1{\cdot}618\frac{a^2}{A^2}}}$$

Where A = area of the pipe.
,, a = area of the orifice.

When the discharge from a vessel is through a short cylindrical tube projecting inwards, the area of the jet contracts more, and the coefficient is reduced to ·5, the velocity being that of a discharge through an orifice. Experiments with short cylindrical tubes projecting outwards (where the length is not less than 2½ times the diameter) have shown that the contraction of the area of the issuing stream is less, and the consequent amount of discharge is greater, than in the case of discharge through an orifice. The velocity of a jet from a short cylindrical tube projecting outwards, has been ascertained to be ·82 of that due to the theoretical head. As the head varies with the square of the velocity, the head due to the velocity of the jet from the tube or pipe will vary as the square of ·82, which gives ·67 as the coefficient for head due to velocity. If A represents the sectional area of the short tube, and D the discharge, the expression is $D = \cdot 82\, A\ \sqrt{2gH}$. The angle at which the particles of water approach the orifice governs the extent of the vena contracta. As the angle of convergence of the water towards the orifice becomes small, the area of the jet approaches more nearly to unity. In the case of a short tube projecting inwards the angle of convergence is great, and the coefficient is ·5.

When the fluid passes into a tube shaped like a cone, having an angle of 20° (this being the natural form assumed by a body or stream of water issuing from a reservoir through a short tube), the coefficient of discharge has been found to be as high as ·95 of the theoretical discharge. As the angle of convergence diminishes, the coefficient also diminishes, until the tube reaches the form of a cylinder. Where the length exceeds the proportion of 2½ times the diameter, experiments have not as yet clearly determined the coefficients.

Bernouilli and Venturi noticed that when the outlet tube from a reservoir was shaped like a truncated cone, with the larger base outwards, the discharge was greater than from a cylindrical tube. The result of various experiments of Venturi indicates that the greatest advantage is obtained when the conical tube has a length nine times the diameter of the

smaller base, and when the angle of convergence is rather more than 5°. This is of great practical importance, and it is frequently overlooked in calculating discharges.

In dealing with the flow of water through long pipes, in addition to the loss of head due to velocity at entry (which applies to the discharge from orifices and short pipes), there is the further loss due to the friction of the water against the side of the pipe. Many experiments have been made to determine a coefficient to represent the loss due to friction. It is obvious that different values will be obtained according to the condition of the surface of the pipe. The observations of Darcy have enabled the following coefficient of friction to be calculated for velocities exceeding 4 inches per second. Where the pipe is a wrought iron or smoothly-coated cast iron pipe—

$$C = \cdot 005\left(1 + \frac{1}{12d}\right)$$

Where the pipe is old and has become slightly incrusted—

$$C = \cdot 01\left(1 + \frac{1}{12d}\right)$$

The relation between the difference of head or level h, and the length of a pipe l, is termed the hydraulic gradient, and the following expression has been determined:—

$$h = \frac{4l}{d} \times \frac{v^2}{2g}$$

In this equation loss by friction is not allowed for, the coefficients for which have been given.

From Eytelwein's experiments a formula is deduced for pipes running full under pressure as follows:—

$$D = 4\cdot 72 \sqrt{\frac{d^5 h}{l}}$$

$$d = \cdot 538 \sqrt[5]{\frac{l\, D^2}{h}}$$

Where D = discharge in cubic feet per minute.
,, d = diameter of pipe in inches.
,, h = head of water in feet.
,, l = length of pipe in feet.

Darcy's and Bazin's formula is—

$$RS = Cv^2$$

Where R = hydraulic mean depth.
S = sine of the inclination (or total fall ÷ total length).
C = coefficient.
v = mean velocity.

The value of C in feet = $\cdot0000457\left(1 + \frac{\cdot0984}{R}\right)$.

Expressing the coefficient in inches (by multiplying by 12^2)—

$$C = \cdot0000457 \times 144\left(1 + \frac{\cdot0984}{R}\right)$$

$$= \cdot0065808\left(1 + \frac{\cdot0984}{R}\right)$$

Mr. Neville has deduced the following general formula to determine velocity—

$$v = 140\sqrt{rS} - 11\sqrt[3]{rS}$$

Where v = velocity in feet per second.
r = hydraulic mean depth in feet.
S = total fall ÷ total length.

The foregoing formulæ apply to pipes of small sizes; observations with large pipes have shown that the formulæ require modification.

Mr. Neville experimented on a 33″ main 11,800 yards long, the gradient of which was 20 feet per mile, or 1 in 264. The discharge was calculated at 15,000,000 or 16,000,000 gallons per day, but in practice it was found capable of conveying above 20,000,000 gallons per day, or 37·36 cubic feet per second, equivalent to a velocity of 6·28 feet per second. Therefore the value of $C = 122{\cdot}96$ in the formula $v = c\sqrt{RS}$.

Mr. Stearns experimented in America on a cast iron pipe 48 inches in diameter, and 1747 feet in length, coated inside and out. The temperature of the water was about 38° Fahrenheit. It had two bends 500 feet and 1170 feet radius.

Mr. Stearns' experiments are given in the following table, reduced by this formula:—

$$v = c\sqrt{R \times S}$$

Where v = velocity in feet per second.
" c = coefficient.
" $R = \frac{1}{4}$ mean diameter of pipe in feet.
" S = inclination.

No. of Experiment.	1.	2.	3.
Quantity in cubic feet per second .	46·972	62·391	77·852
Mean velocity	3·738	4·965	6·195
Total loss of head	1·243	2·133	3·230
Gradient	1 in 1405	1 in 856	1 in 540
Value of "c" in $v = c\sqrt{RS}$.	140·14	142·11	144·09

These results are about 25 per cent. higher than by Darcy's formula.

All the foregoing formulæ are based on the friction varying with v^2. Hagen, however, pointed out as far back as 1854 that this was not the case, and Dr. Lampe in 1873 came to the same conclusion. Professor Osborne Reynolds in 1883 brought before the Royal Society the results of a series of observations he had made, which led him to conclude that, instead of the friction varying as v^2, it varied as $v^{1\cdot722}$ with lead pipes, and as $v^{1\cdot7}$ for the smoothest pipes, reaching v^2 where the surfaces were roughest. It follows, therefore, that a formula should be employed for determining the discharge which avoids the inconvenience of using a coefficient that varies with the velocity. Professor Unwin deduces a formula from Dr. Lampe's data as follows:—

$$\frac{h}{l} = m\frac{v^n}{d^{3-n}}$$

Where the values of m and n are as follows:—

	Diameter of Pipe.	m.	n.
Lampe	1·373 feet	·0003707	1·85
Darcy	·6186 ,,	·000379	1·95
Darcy	1·643 ,,	·000324	1·95

Mr. Mair has recorded a series of experiments made by him

on a $1\frac{1}{2}$-inch brass pipe, which show that the value of m is affected by temperature to the following extent:—

Fahrenheit. Degrees.	m.
57	0·000276
70	0·000263
80	0·000257
90	0·000250
100	0·000244
110	0·000235
120	0·000229
130	0·000225
160	0·000206

The value of n in this case was 1·795.

The formula would therefore require modification to meet variations of temperature, and would be as follows:—

$$\frac{h}{l} = \cdot 00031 \left(1 - \cdot 00215t\right) \frac{v^n}{d^{3-n}}$$

Where t = temperature in degrees Fahrenheit.

Mr. Edgar Thrupp (one of the author's assistants) has made a careful examination of the experiments of Darcy, Reynolds, Lampe, Hamilton-Smith, jun., Stearns, Leslie, Couplet and others, and his results indicate that the discharge does not vary as $D^{2\cdot 5}$, as has been previously assumed, but more nearly as $D^{2\cdot 6116}$, and that this index is apparently independent of the nature of the pipe.

The following formula is based on the fact that the friction does not always vary as v^2, and also obviates the necessity of applying different coefficients to pipes of different diameters, which is necessary in all the foregoing formulæ—

$$Q = \frac{D^{2\cdot 6116}}{c \sqrt[n]{S}}$$

Where Q = discharge in cubic feet per second.

D = diameter of pipe in inches.

S = cosecant of inclination = $\frac{\text{length.}}{\text{head.}}$

$\left.\begin{matrix} n \\ c \end{matrix}\right\}$ = constants depending on the nature of the pipe.

For new cast iron—

$n = 2{\cdot}00 \quad c = 13{\cdot}66$ (lowest).
$n = 2{\cdot}00 \quad c = 15{\cdot}00$ (average).
$n = 2{\cdot}00 \quad c =$ about 16 (highest).

Also—

$n = 1{\cdot}78^* \quad c =$ about 10.

For well-cleaned old cast iron—

$n = 1{\cdot}75 \quad c = 11{\cdot}74.$

For old incrusted iron—

$n = 2{\cdot}00 \quad c = 22{\cdot}5.$

For wrought iron gas piping—

$n = 1{\cdot}90 \quad c = 11{\cdot}5.$
to $n = 1{\cdot}85 \quad c = 16{\cdot}6.$

For wrought iron pipes generally—

$n = 1{\cdot}85 \quad c = 10{\cdot}7$ to $14{\cdot}5.$

For lead pipes—

$n = 1{\cdot}70$ to $1{\cdot}78 \quad c = 9{\cdot}33$ to $10{\cdot}7.$

Average—

$n = 1{\cdot}72$ and $c = 10.$

The formula does not apply below the "critical point" noted by Professor Osborne Reynolds, and referred to hereafter. The position of this critical point may be roughly indicated as follows, at which point the formula will break down:—

When $S = 10{,}000$ and Q is less than $\cdot0060$
$S = 1{,}000$ „ Q „ $\cdot0018$
$S = 100$ „ Q „ $\cdot0010$

If extreme accuracy is desired, the values of "c" must be modified for temperature, but this will seldom be necessary in practice.

It may be noticed that the experiments of past and present observers, from which the various formulæ have been deduced, have all been with water at comparatively low pressures. No

* Where two values of n are given, the discharge should be worked out for each with its accompanying value of c. The one which gives the lowest discharge is the correct one to accept. The change in the value of n from $2{\cdot}00$ to $1{\cdot}78$ is indicated by Darcy's experiments, series xvi., xvii., and xviii.

similar observations have been made with water at high pressures. Experience so far seems to prove that friction in water-mains is practically independent of pressure. As the loss by friction is dependent on velocity and not on pressure, in the practical application of water to the purpose of actuating machines, the higher the water pressure is, the less does the element of friction in the pipe come into account. Gauges have been placed on a high pressure main composed of 4-inch, 3-inch, and 2-inch pipes in the Great Western Railway-yard at Paddington, at points from 1000 to 1600 yards apart, and the pressure has been found to be practically the same during the working of the machines in the usual way. At the Swansea Docks, wherever the pressure in the main has been tried, it has been found to be uniform.

When the pipe ceases to have a uniform internal diameter, important fluctuations of velocity and pressure arise, which are too frequently forgotten. A sudden enlargement or reduction in a pipe produces eddies in the current, which result in a corresponding diminution or increase in the velocity, and therefore in the pressure. A vein of water flowing at a uniform velocity is influenced throughout its mass by a uniform pressure acting upon it from behind. If this vein suddenly reaches an enlargement in the pipe, the velocity will be diminished, as the velocity in a pipe varies inversely with the area. The reduced velocity through the enlarged portion of the pipe implies that a force is opposing the forward movement of the water; or in other words, the force from behind (which produced the forward movement) meeting a force in front which arrests or diminishes it, implies that a pressure must be produced in front of the fluid increased beyond that which existed at the time it was flowing in the uniform portion of the pipe. Similarly, when a vein of water meets a contraction in a pipe, the diminished sectional area of the pipe necessitates an accelerated velocity proportional to the reduced area. This shows that the pressure behind the fluid is greater than that in front, and consequently the pressure throughout the length of pipe

which has the reduced area is less than it was before, in proportion to the extent of the reduction.

It follows, therefore, that in a system of pipes of varying areas, in which a fluid is circulating, the pressure that is exerted by that fluid will vary at any point in inverse proportion to the velocity at that point, or in other words, to the sectional area at that point. In the lengths of pipes where the sectional areas are the same, there will be found the same pressures (friction not being considered). Where the areas are greatest, there will

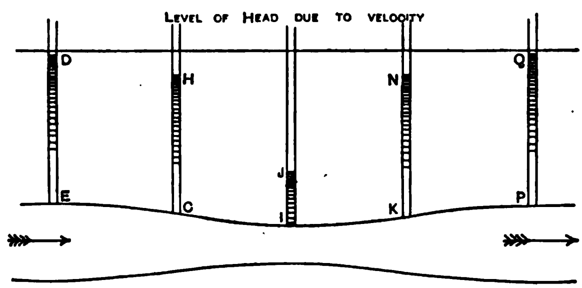

FIG. 1.

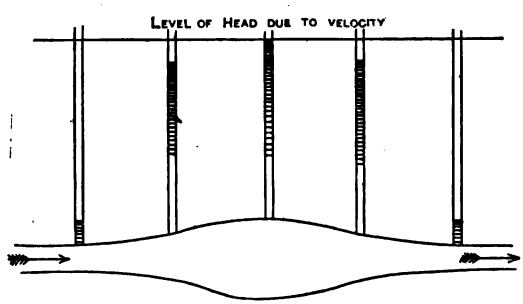

FIG. 2.

be the greatest pressures, and where the areas are least, there will be the least pressures. Experiments by Mr. Froude proved this. He inserted a series of vertical glass tubes in a horizontal length of pipes which had enlargements and contractions in them as shown by figs. 1 and 2. It will be seen that the heights to which the fluid rises when it is flowing through the pipes will vary (with a uniform head) as the area of the pipe varies. The tube which is placed over an enlargement in the

pipe will have a higher column standing in it than is the case with the tube which is placed over a contracted length. For instance, on fig. 1 the sectional areas at E and P are the same, and the areas at C and K are also equal to each other, but are smaller than those at E and P, whilst the area at I is the smallest of all. Disregarding the slight loss of head due to friction, the level to which the fluid rises at the various points is indicated by the heights ED and PQ, CH and KN, and IJ. Fig. 2 shows similar results, equal sectional areas producing equal pressures, and consequently equal heights. In other words, a pipe filled with water, and subjected to a head, has a greater pressure in it when the water is at rest than when the water is in motion; again, it has a greater pressure when the velocity is checked by an enlargement in the pipe, and has a less pressure when the velocity is accelerated by a reduction in the pipe. This has been expressed by Professor Cotterill thus:—If $Z =$ the elevation or actual head for any section of the pipe, $p =$ the pressure, $u =$ the velocity, $W =$ weight of a cubic foot of water, then

$$\frac{u^2}{2g} + \frac{p}{W} + Z = \text{constant},$$

and he describes the first term of this equation as representing "energy of motion," the second "energy due to pressure," the third "energy of position," the total energy of the water thus being shown to remain constant as it traverses the pipe (frictional resistance not being considered). If the friction of the fluid be taken into consideration, some of the head would be lost, and this loss produces what is termed the hydraulic gradient, as is shown by fig. 3.

Where a current of water is passing at a high velocity from a large pipe through a small opening or valve, a loss of useful effect follows. In the case of water flowing through a contraction (such as it meets with in valves), there is a practical loss of the head which existed in the main before the water entered the valves on the way to the cylinder. The head which induced the acceleration of velocity does not all re-appear upon the

cylinder side of the valve, and do useful work in the machine. It has produced velocity at the expense of head. The potential

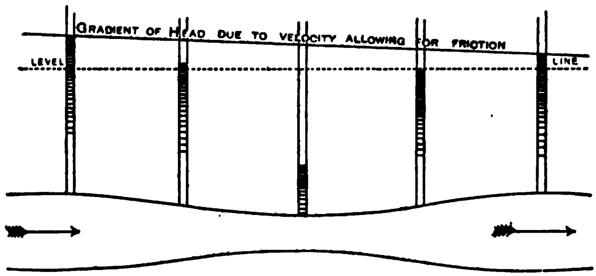

FIG. 3.

energy thus lost is equivalent to the product of the weight of water and the height it would have to fall in order to acquire that velocity. The loss of pressure energy occasioned can be stated in terms of the respective velocities. If v represents the accelerated velocity, and v' the original velocity before the contraction, then

$$\frac{(v - v')^2}{2g}$$

will be the loss of energy sustained by the interposition of the contraction. On the water entering the cylinder of an hydraulic machine, the pressure which it would be capable of exerting would be in proportion to the velocity it has in the cylinder, less the loss due to eddies. The change of direction of a portion of the water, from a straight line to a curve or eddy, involves the loss of the energy which is exerted in producing the rotative action of the particles of the water in the eddy, and a corresponding diminution of the energy which produces the forward flow of the water. Eddies are formed by a volume of water at one velocity mixing with water either at rest or at a different velocity, motion being induced, involving a loss of the head which is necessary to produce it. The formation of eddies in a line of pipes, or in machines utilising water as the motive-power, is therefore attended with a loss of effective power. The loss of head due to the passage of water through valves, or other sudden contractions, where eddies and shocks are pro-

duced, is not capable of being reduced to a simple formula, as the element of friction prevents the direct relation between velocity and pressure from being relied on, as in the case of simple enlargements and contractions in tubes.

Weisbach experimented in this direction, and arrived at the following coefficients:—When a tube of uniform section is contracted by a diaphragm, the original pressure is reduced from 1 to ·624 when the tube is contracted to $\frac{1}{10}$th of its original area, and from 1 to ·681 when contracted to one-half.

From what has been already stated, the difficulty will be realised of endeavouring to express (in any formula worth adoption by those engaged in the practical construction of hydraulic appliances) the loss of energy produced by the passage of water under pressure through valves or cocks under any of the usual conditions. It is important that the supply and exhaust valves should be large, and that they should open quickly (compared, that is, with the ports of a steam cylinder). The efficiency of many hydraulic appliances is diminished by the ports being unnecessarily contracted, and by sharp bends being introduced.

It has been generally considered that if a body of water is passing through a pipe in lines or threads parallel to each other, it will continue to do so, provided no change of shape or interruption is caused in the pipe. Professor Osborne Reynolds has however shown (in the paper already referred to), that beyond a certain velocity (which he terms "critical velocity") the fluid ceases to flow in parallel lines, but suddenly bursts into eddies, a viscous fluid being less liable to form eddies than a non-viscous fluid, and an increase in temperature increasing the tendency to form eddies. This change of a steady motion into an unstable or sinuous motion is of the greatest interest and importance. Previous experiments had been confined to the motion of water in a pipe under the two conditions in which the resistance varied either as the velocity or as the square of the velocity. No apparent definition had been made of the point at which this change of law occurred,

or as to the circumstances which produced the change from steady to unsteady motion, that is, from motion in parallel lines to motion in sinuous or eddying lines.

Professor Reynold's experiments were made with glass tubes about 4 feet 6 inches in length, and 1-inch, ½-inch, ¼-inch diameter, with trumpet-shaped mouths. They were arranged so as to be able to draw water out of a large glass tank in which they were immersed, whilst a streak of coloured water was admitted at the point of inflow of water into the pipe. Fig. 4

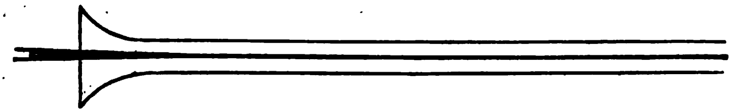

FIG. 4.

shows the result when the velocities were low, the coloured streak continuing in a straight line. As the velocity was increased, a point was reached when the coloured streak would suddenly break up and become mixed with the clear water, as shown by fig. 5. As the velocity increased, the point at which

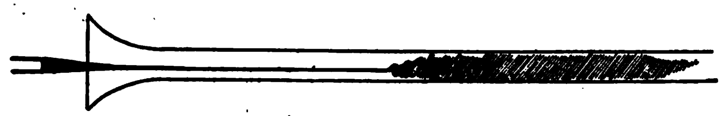

FIG. 5.

the break-up occurred approached the trumpet mouth. By the aid of the electric spark, the eddying or curling appearance of the coloured water was made apparent, as shown in fig. 6. It

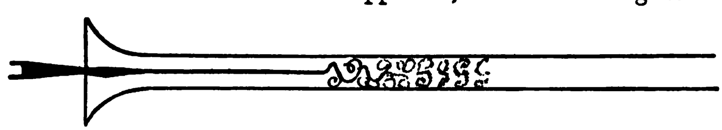

FIG. 6.

was found that the velocity at which the junction of eddies occurred, was almost exactly in the inverse ratio of the diameter of the tubes, and that the critical velocity diminished as the temperature rose.

Further experiments were made with water flowing in two straight lead pipes, each 16 feet long and ½ inch and ¼ inch in

diameter. Gauge holes were made at 10 feet and 15 feet in these pipes, and the results proved that at lower velocities the pressure was proportional to the velocity, and that the velocities at which a deviation from the law first occurred were in exact inverse ratio of the diameters of the pipes. Also, that when a velocity equal to the critical velocity multiplied by 1·2 was reached, the pressure did not vary as the square of the velocity, but as 1·722 power of the velocity, as already stated.

GENERAL OBSERVATIONS.

The facility with which water under pressure is capable of being transmitted, and the advantages that attend its utilisation in motors, have resulted in many practical difficulties being overcome which were at first encountered. Bramah's anticipations have been thus realised, as he had before his mind, and foreshadowed, the great opening that existed for an extended use of water-power, although a long interval elapsed, after Bramah's labours had ended, before Sir William Armstrong devoted himself to the subject of utilising water-pressure. He was led to its consideration from a standpoint of his own, independently of Bramah's previous efforts. To the utilisation of the enormous natural stores of water-power which ran to waste in the rapid falls of streams and rivers, he, at the outset, directed his attention, as being both desirable and feasible. He began by experimenting upon the application of water-power to actuate a rotating machine by forcing a series of pistons through a cylinder. He then applied the low pressure of the town main at Newcastle (which had an effective head of only about 200 feet) to working a crane by means of rams and sheaves. A greater pressure was next obtained by pumping into a high level tank, and ultimately the Accumulator was constructed which is described hereafter.

The pressure that has been adopted for transmission through mains, in order to work ordinary hydraulic machines, is 700 lbs. per square inch. This is found to be a convenient work-

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ing pressure, both as regards the size and proportions of the working parts and the tightness of joints and valves. Where hydraulic power, however, is applied to shop tools, the pressure mostly used is 1500 to 2000 lbs. per square inch. In some applications of water-power, the pressure is even carried up to 15,000 lbs. per square inch.

The low pressure of a waterworks main can be used to actuate hydraulic appliances with advantage and convenience. More than twenty-five years ago the author (at that time on Sir William Armstrong's staff) erected a hydraulic crane and lift at the Chelsea Waterworks Station on the Thames at Surbiton (Mr. James Simpson, Past. Pres. Inst. C. E., being the engineer). The power to work these was derived from the low pressure main of the Chelsea Water Company.

At the outset of the employment of water-power, it was feared that the water in the pipes and machinery might freeze. This, however, has been found not to be a difficulty where well-known precautions are taken. The working parts should, where possible, be placed under ground, or should be cased in, if they are above ground. The water should be run out of all valves and cylinders which cannot be cased in, and protected as soon as the working of the machine ceases. A very small gas jet or lamp placed near the unprotected parts will prevent freezing.

Experiments have also shown that a mixture of glycerine and water prevents the effects of frost to a temperature as low as 16° Fahr., provided the glycerine has a specific gravity of 1·125, and that it is mixed in the proportion of one part of glycerine by weight to four parts of water. Where water is scarce and is used over again in the machines (by returning the exhaust water from the machines to a reservoir), such addition of glycerine is more easily resorted to. Where moderate risks of frost have to be dealt with, the proportion of one gallon of glycerine to 300 gallons of water proves effectual. If the water is at a high pressure, such as 1500 lbs. to the square inch, it is less liable to freeze than when it is at a low pressure.

Again, it was at first feared that accidents would be frequent

from the bursting of hydraulic pipes and cylinders under high pressure. Such, however, has been proved not to be the case in practice, and even where pipes or cylinders do burst, the pressure is at once dissipated, as the body of water which can issue at the opening is but slight.

The best way of jointing hydraulic pipes has been the subject of much practical experiment. A guttapercha ring has been universally adopted as the best means of preserving the joint watertight. Fig. 7 shows the usual high pressure flange joint.

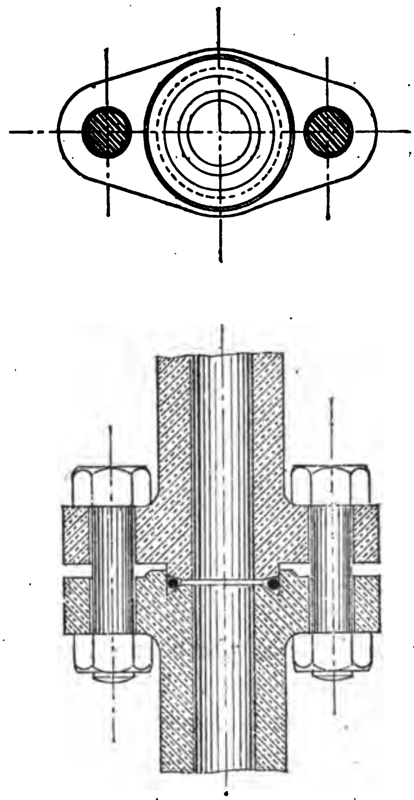


Fig. 7.

Mr. Ellington has designed a modified form of this joint by putting a projection on the pipe beyond the flange, the spigot and faucet being formed on this projection. The effect is to increase the depth and the strength of the flange, without an increase of its section at the junction between the flange and the pipe.

If the thickness of the metal between the body of the pipe and the flange is not skilfully proportioned, the strength of the flange becomes weakened by unequal cooling.

In laying hydraulic mains under public roads where they will be subject to vibration owing to heavy traffic, their thickness and strength should be made greater than usual, and if the ground is not good, concrete should be placed under the

pipes. In ordinary cases, where the main is not likely to be exposed to any strain or pressure except that due to the head, the working pressure would be calculated at about one-fifth of the bursting pressure. A less proportion, such as one-eighth, should be adopted under less favourable conditions.

Where hydraulic power is applied to movable machines, the continuity of the supply to meet the varying positions of the machines is insured by a ball and socket joint, and by a double right-angled joint, as shown by fig. 8.

As water is non-elastic, provision has to be made against the effect of suddenly arresting the momentum of the load acting

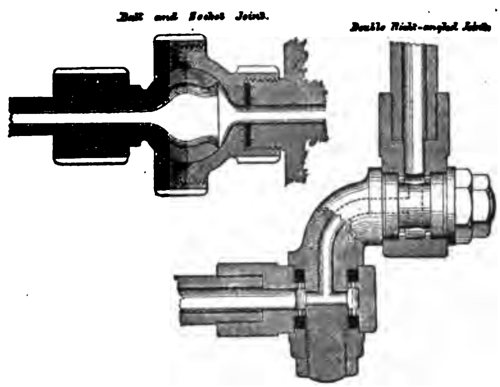


FIG. 8.

upon the ram, piston, or other working parts of a quick-acting hydraulic machine. This is accomplished by introducing relief valves. Fig. 9 shows their application to the slide valve of a crane. P, P represent the pressure, and E, E the exhaust passages. It will be seen that the small relief clacks, D, D, open against the pressure in the supply pipe P, and absorb any blow or concussive action in the cylinder or passages, which raises the pressure above that of the accumulator.

It is desirable to use water which is as free as possible either from suspended matter or from chemical impurity. The former increases the wear and tear of the packing, and is otherwise inconvenient, and the latter acts injuriously on the seats

and fittings of valves. Sea-water can be used for hydraulic machinery, although fresh-water is better.

Water-pressure has sometimes been applied to actuate machines which are worked continuously and not intermittently, and to continuous working rotary machines. This is unwise, for in applying hydraulic power to the continuous working of shafting or shop tools, the amount of power developed by the hydraulic engine cannot be varied to suit the work to be done, neither can the speed be regulated with sufficient nicety.

There is ample scope for the economical employment of water-power without adopting hydraulic pressure as the means of working apparatus which is to be in continuous operation.

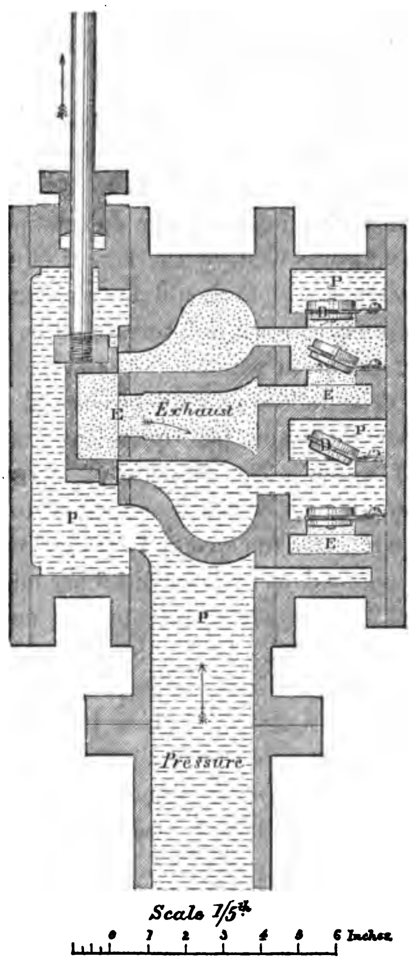


FIG. 9.

The mechanical energy that is derivable from water flowing in a natural stream can be applied either in the form of a weight of water falling through a height, or in the form of velocity due to previously acquired motion, in addition to that due to its falling through a height. It is also derived from pressure due to a column of water in a pipe subject to the head of water in a stream. The power which exists in natural streams and falls of water is to a very great extent wasted. Its utilisa-

tion is possible by a variety of water-pressure machines which afford a means of conserving, distributing, and thus rendering available this energy. By converting the water-power (due to either head or velocity) into rotary motion, it can be stored in accumulators, at or near the point of production, and can be distributed at high pressures through small pipes to distant parts, where its energy may be employed usefully.

If H represents the difference in the level of the water between the point where it is impounded for utilisation in a pipe, and the point where it is to be applied to actuate a machine, then a pound of water descending through H feet has communicated to it, by the action of gravity, an energy during its descent which should be represented by the term "H foot" pounds of work, and this would be available for working motors at the lower points.

The following data are useful in calculating power:—One pound pressure per square inch is equivalent to 2·3 feet head of water. The pressure of the atmosphere is 14·7 lbs. per square inch, which is equivalent to 34 feet head of water. As a gallon of water weighs 10 lbs., a flow of one cubic foot per second is equivalent to 375 gallons per minute. The unit of power is 33,000 lbs. falling one foot in one minute, which is a horse power (HP). As a cubic foot of water weighs 62·2 lbs., one HP is 530·5 cubic feet of water falling one foot per minute. This represents the gross HP, but the actual HP is less than this, depending on the efficiency of the prime mover. If 74 per cent. were to be taken as the coefficient of useful effect, 1 HP would be obtained from 720 cubic feet of water falling a foot per minute.

WATERWHEELS.

In 1838 Sir William Armstrong devised a motor (described in the *Mechanics' Magazine* at the time) which combined the use of pistons forced by water-pressure through a cylinder,

with the continuous rotation of a waterwheel. The machine consisted of a waterwheel with a flat rim containing four equidistant pistons, folding into circular apertures. These intersected longitudinally a curved tube open at the lower end, and communicating at the upper end with the supply pipe. The pistons opened out as they entered the tube, and folded up as they left it, each piston taking up the pressure before the preceding one lost it. One of these was worked at Newcastle and Gateshead in 1839 with a head of 131 feet of water, when it rotated at thirty revolutions per minute (the limited supply of water preventing a greater speed), and it developed 5 HP with a coefficient of 95 per cent. of the theoretic power of the fall.

A balanced float-water-motor recently devised by Mr. Sealey Allin was shown at the Exhibition of Inventions (1885). It consists of a series of feathering floats hinged to a chain working over two drums, one at a high level, where the head of water is applied, and the other at the low level, corresponding to the race of a waterwheel. These floats descend through an enclosed case with a clearance of $\frac{1}{64}$th of an inch, by which leakage and friction are avoided. The feathering of the floats is automatic, being regulated by the level of the tail water, the series of floats moving forward when the pressure behind exceeds that in front. The top drum has a series of projecting teeth, which take the links, and so convert the power into rotary motion. The speed of the drum is about 180 revolutions per minute, and the efficiency upwards of 90 per cent.

In considering the application of water to wheels, the variation of level of the water determines the character of the wheel to be employed. The rise in the stream, and in the level of the water for a given width of wheel, is sometimes greater than can be utilised in the open bucket of an overshot wheel. A variation of level of 2 feet, and a velocity of the periphery of the wheel of 5 feet per second, are the usual limits. When the water is received by the wheel below the summit (generally between the axis and the lowest point), it is called a "Breast

Wheel." In this case the supply is regulated by an adjustable sluice, over which the water flows to vanes on the wheel, which are substituted for buckets. A channel of brickwork or stone surrounds the wheel, and in this the vanes move. The water, filling this channel, acts by its weight in turning the wheel. The efficiency of an overshot or breast wheel sometimes reaches 75 per cent., but a lower efficiency of about 65 per cent. is a safer estimate, unless the wheel is very well designed and constructed.

In the ordinary overshot wheel, the open buckets of wood or iron are shaped so as to prevent the water shooting over the wheel. This is also obviated by making the capacity of the bucket sufficient to hold about three times the volume of water discharging into it. The buckets are placed between two shrouds, and the power exerted by the wheel is measured by the volume of water, and the height through which it falls. If H is the net fall in feet, Q the weight of water per second in pounds, then the gross horse-power exerted by a wheel will be—

$$\frac{Q.H.}{33{,}000}$$

Besides the power due to head, there is the additional power due to velocity,

$$\left(\frac{v^2}{2g}\right)$$

when the water is delivered to the wheel with a previously acquired velocity.

If V be the velocity which the water has acquired before it is discharged on the wheel, v the diminished velocity after it has been acting on the wheel, and Q the weight in pounds of water which has been discharged per second, then the theoretical power exerted on the wheel will be—

$$\frac{Q\,(V-v)^2}{2g}$$

and the power of the stream will be—

$$\frac{Q. V^2}{2g}$$

If h be the head due to velocity, and acting on the vanes of the wheel, the speed with which the wheel should revolve in order to produce the greatest efficiency is determined by taking four times the square root of h, or rather less. The water loses altogether about half of the useful effect due to this velocity, as it moves on with the wheel after it has impinged upon it, and further, because some of the force has been dissipated in coming into contact with the vanes. The vanes should be shaped so that the water on entering them does not suffer loss due to change of direction. The water should leave the wheel with the lowest velocity possible.

In the "Undershot" waterwheel the water acts by its velocity at the bottom of the wheel. When small falls of six feet or so, or rapid currents, have to be utilised, the undershot waterwheel, as perfected by General Poncelet, is a most efficient prime mover. Beyond that fall the wheel has to be large and costly to give much power, and a "Turbine" is preferable. Even when the undershot waterwheel is working under favourable conditions, at least one-half of the energy due to the fall is lost. The water impinging on the floats imparts some of its energy to the wheel, but it loses part through eddies and breaking up. After it has acted on the floats, again, it passes away from under the wheel with a velocity at least equal to that of the wheel, all of which velocity represents lost energy. About 25 per cent. of loss may be attributed to each of these causes, although in good undershot wheels 60 per cent. of efficiency is possible.

Poncelet adopted the following rules as enabling the best results to be obtained:—The water should impinge on the curved buckets at the bottom of the wheel at an inclination of 1 in 10. The diameter of the wheel should be twice the depth of the fall. The velocity of the periphery of the wheel should be arranged to be 55 per cent. of the velocity due to the head, measured to the centre of inlet. The fall and volume of water

being known, the power of a wheel is determined thus, taking 60 per cent. of efficiency:—

Q = Volume of water in cubic feet per minute.
HP = Effective horse-power.
F = Fall in feet.
530·5 = cubic feet of water per minute falling 1 foot = theoretical horse-power.
c = coefficient of useful effect.

Then

$$HP = \frac{Q \times F \times c}{530 \cdot 5}$$

If

Q = 3000 cubic feet per minute.
F = 4 feet.
c = 60 per cent.

$$HP = \frac{3000 \times 4 \times \cdot 6}{530 \cdot 5} = 13\tfrac{1}{2}$$

In America the "Pelton" waterwheel is reported to have an efficiency as high as 80 per cent., owing to the substitution of cups for flat floats. The water is delivered to the wheels through nozzles of from 1 inch to 2 inches diameter, and the water impinges on the wheel like a jet striking a hollow cup. The water is thus not broken up, but spreads and exhausts the whole of its energy in the cup.

TURBINES.

In the old "Barker's Mill" or "Reaction Wheel," water passes downwards through a vertical tube, which forms the axis of a horizontal tube, having holes at its extremities through which the water issues. This produces a rotative action, but it also causes the water to have a rotary velocity after leaving the tubes of the reaction machine, which involves a loss of power. This attracted the attention of Fourneyron, and led to his inventing the "Turbine." By means of guide blades fixed in an external case, he gave the water a forward motion before it entered the wheel or internal case which revolved on the axis

of the machine. This resulted in the water passing out of the machine at right angles to the axis, without a backward velocity, thus avoiding the corresponding loss of energy. The machine devised by Fourneyron forms essentially the basis of the numerous turbines which have been subsequently invented by Jonval, Professor Thomson, Schiele, Girard, and others. In well-constructed turbines, the loss of energy due to velocity after the water leaves the machine varies from 5 to 8 per cent. The loss from skin friction depends on the size and form of machine.

A series of experiments by Mr. Lehmann on thirty-six turbines, varying from 1 to 500 HP, led to the following estimate of losses:—

Loss per cent. due to	Axial Flow. Turbine.	Outward Flow. Turbine.	Inward Flow. Turbine.
Hydraulic resistances	12	14	10
Unutilised energy	3	7	6
Shaft friction	3		2
Total	18	23	18
Efficiency	0·82	0·77	0·82

In ordinary practice, the efficiency of a turbine should not be taken at more than from 75 to 80 per cent., although it is claimed that a higher efficiency has been obtained in some turbines.

The power of a turbine can be calculated by the following formula:—

HP = effective horse-power.
Q = Volume of water expended in cubic feet per second.
F = Head or fall of water acting on turbine.
c = coefficient of useful effect.

$HP = Q \times F \times$ (weight of a cubic foot of water) $\times$ 60 secs. $\times\, c$.
If $c = 75$ per cent.
$HP = {\cdot}085\ Q \times F$.

CENTRIFUGAL PUMPS.

For raising large quantities of water a small height, a "Centrifugal Pump" (which is practically an inverted turbine) is a very suitable form of pump. Appold constructed the first, and it is practically the basis of all subsequent ones. In this form of motor it is necessary to bear in mind that the greatest efficiency can be only obtained when it is applied to work under a constant head. The calculations on which the shape and design of the motor are based, show that an equally good result cannot be obtained when the head is variable. A velocity of about 5 feet per second for the flow of the suction and discharge water is generally regarded as that which should be aimed at. The disc friction varies as the square of the diameter, and the loss due to total frictions increases as the cube of the velocity. Experiments with centrifugal pumps have established an efficiency of about 50 per cent. in the small pumps, and about 70 per cent. in the large pumps. The shape of the curved vanes of the fan materially affects the results, the best form being that in which these are bent backwards.

Professor Unwin has shown by his experiments on the "Friction of Discs Rotated in Fluid" (recorded in the *Proceedings of the Institution of Civil Engineers*) the conditions which require to be observed in order to minimise the loss of efficiency in turbines and centrifugal pumps. This loss is largely due to the friction of their disc-shaped surfaces in the water surrounding them. The larger the chamber in which the disc rotates, the greater is the friction of the disc, which is attributable to the stilling of the eddies by the surface of the stationary chamber. The stilled water reacting upon the surface of the disc causes the friction of the disc to be dependent, not only on its own surface, but also on the surface of the chamber in which it rotates.

By reversing the action of inward flow machines (by which water is forced upwards through the centre), the machine be-

comes an outward flow pump capable of raising water the same height as the fall, after deducting loss from friction.

A "Chinese" (or "Scoop") wheel is simply an overshot water-wheel with reversed motion, by which water is caught up in the buckets of the wheel as it revolves, and is raised to a height nearly equivalent to the diameter of the wheel. For low lifts of about 10 feet, and for large. volumes of water, this form of pump has an efficiency of upwards of 80 per cent. The diameter is usually from four to five times the height of the lift, and the speed of the periphery should be about 8 feet per second. This kind of motor is adapted to the draining of fen lands.

Mr. Wilfrid Airy has designed an "Archimedean Screw" pump for lifting fluids, which illustrates the great efficiency obtainable from a motor which is designed to avoid loss of energy from eddies or shocks in the translation of the fluid. A description of this pump was given to the Institution of Civil Engineers in 1871. It consists of a rotating cylinder, having a central core, and one or more spiral passages. It works in a frame at an angle of 30° to 45° with the horizon, and the velocity of rotation is about 3 feet per second, measured on the periphery of the cylinder. The lower end is placed in the water to be raised, and the upper end is attached to the delivery. In the working of a well-constructed Archimedean screw pump on this plan, an efficiency as high as 85 per cent. is obtained. The feature in this motor is that the spiral threads are made on the "developable" system, or that by which a curved surface can be unwrapped, laid flat, and made into a plane.

WATER-PRESSURE PUMPS.

The single-acting lift pump delivers intermittently by means of a pump bucket. The single-acting plunger pump differs from the former, the bucket being replaced by a ram. Water is raised into the pump barrel through a foot valve during the ascent

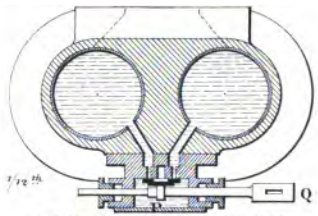
of the ram, and is forced through a delivery valve during the descent of the ram. It is thus intermittent in its delivery. By a duplication of the valves it is converted into a double-acting pump, the water being drawn down into, and forced out of, the barrel continuously, the valves being actuated by the water itself.

Trevithick applied water-pressure direct to a pump to raise water from mines in Cornwall. In this early form of pump, the water was admitted alternately to the upper and lower portions of a piston working in a pump barrel, and controlled by valves actuated by tappets on the piston rod. The reversal of the valve at the end of each stroke caused the sudden stoppage of the flow of water, and consequently produced shocks, which were mitigated by allowing a little water from the column which worked the engine to pass through the valve each time, so that the column was not wholly stopped. A corresponding loss of useful effect, however, was produced.

In applying a natural head of water to work a direct acting pumping engine, the difficulty is to convert the flow of water in a pipe into a smooth-working reciprocating action. If the volume of water to be utilised is large, and the head is small, the velocity of flow should not exceed 10 feet a second, increasing to 20 feet a second with high pressures. An engine worked by water-pressure on the double acting principle is constructed similarly to steam-engines, but with the ports greatly increased in size.

At the Institution of Mechanical Engineers in 1880, Mr. Davey described a water-pressure pumping-engine (shown by Plate 1) which had been designed by him for the Knoll Colliery, Nuneaton. It was used to clear the water from a long working to the dip at an inclination of 1 in 6. The engines were specially designed to be very portable, and to occupy a very small space. They were in duplicate, each having a power cylinder 6¼ inches diameter, and 2 feet 6 inches stroke, working direct on a pump plunger 8¾ inches diameter, working at 12 double strokes per minute. By means of the liner M the diameter of the cylinder can be increased. When the liner is in position (as shown) the

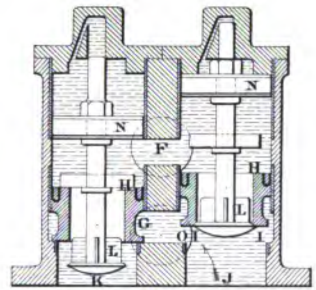
Fig: 2.



Scale 1/12th

PLAN OF ENGINE VALVES.

Fig: 3.



SECTION OF ENGINE VALVES.

Fig: 4.

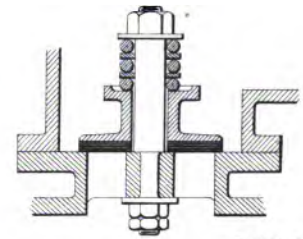
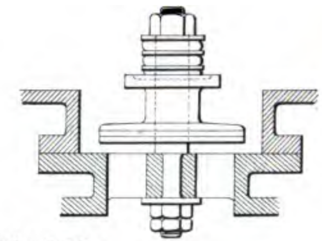


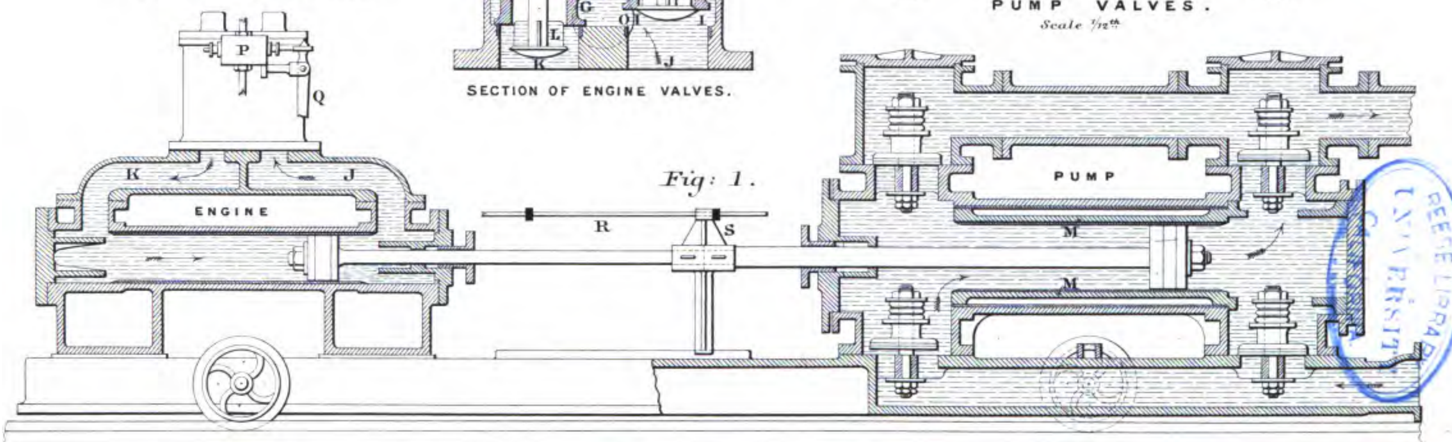
Fig: 5.



PUMP VALVES.

Scale 1/12th

Fig: 1.



LONGITUDINAL SECTION.

Scale 1/24th

Inches 12 6 0 1 2 3 4 5 Feet

Thos Kell & Son, Lith.

pump has a diameter of 8¾ inches. When it is withdrawn, a piston with a diameter of 12½ inches can be employed, by which the capacity of the pump is doubled. The engine can then pump double the quantity of water to half the previous height, which is the arrangement employed until the water is cleared to half the depth of the workings. The water-pressure is supplied to the engine under an effective head of 450 feet, and it is transmitted through about 2000 feet of 5-inch pipe. Each engine pumps 150 gallons per minute to a height of 150 feet, through 800 feet of 7½-inch pipe. The engine and pumps are placed on wheels, by which their position can be altered with respect to the water level.

The valves are shown by figs. 2 and 3. The top orifice F (fig. 3) is the inlet, and the bottom orifice G is the outlet. The pipes J and K form the communication to the two ends of the engine-cylinder. The outlet valves HH are annular gun metal pistons working vertically, each having two valve-beats, one on the inner edge I, and one on the outer edge O, of its bottom face. As this valve descends, the outside beat closes the communication to the outlet pipe, whilst the inlet valve L (rising against the inner beat) closes the supply. The inlet valve is an ordinary single beat mushroom valve with its spindle projecting, and attached to a piston N. The bottom face of this is constantly under the head in the pressure pipe, whilst the top face is exposed alternately to the head in the pressure pipe, and to the pressure in the outlet pipe, by means of a small gun metal slide valve P (fig. 2), actuated by a lever Q, and tappet rod R (fig. 1). If the exhaust valve is closed, and the pressure valve is opened (as on the left-hand side of fig. 3), then the pressure valve L in closing rises up against the annular exhaust valve H, and lifts it, opening the exhaust orifice G. The valves are now in the position shown on the right-hand side of fig. 3. The arm S (attached to the crosshead of the engine) strikes the tappet towards the end of the stroke, and pushes the slide valve over, by which the top of the right hand valve piston is open to the head in the pressure pipe, and that of the other to the outlet.

The main valves are thus reversed; the right hand piston being under equal pressure top and bottom, the pressure on the top of the annular valve H forces it downwards, carrying the pressure valve L with it. When the valve H has come down on its beat O (closing the exhaust orifice), the valve L continues to descend under the pressure above it, and opens K to the pressure, as on the left-hand side of fig. 3. On the other side the ascending piston causes its inlet valve to close and its outlet valve to open. By this arrangement water cannot pass through the valves unless it has performed its work, as the exhaust and pressure valves can never be open at once. The important avoidance of a sudden arrest of the flow of pressure water is thus effected. It was found that the efficiency of the engine was 65 per cent.

Where the water-pressure is great, or where the water is dirty, plunger valves are used, as any form of slide valve, or piston valve, is open to objection. If slide valves have to work under high pressures or in dirty water, a good combination of materials is found in lignum vitæ slides working on brass faces.

Herr Pfaehler employs water at the Sulzbach Altenwald Colliery, near Saarbrücken, to transmit the power from a steam-engine at the surface to actuate pumps at the bottom of a shaft 306 yards deep. The steam-engine has a cylinder 53 inches in diameter, and 61·5 inches stroke, connected with pressure plungers 9 inches in diameter and the same stroke. These plungers are brought into connection with an underground pumping-engine, consisting of four pressure pumps, with plungers 6 inches in diameter and 66 inches stroke, arranged in pairs, and put in motion alternately by the surface plungers. Between each pair of plungers (which are connected by a cross-head) is placed the working plunger of one of the mine pumps. The engine at the surface transmits the effort of each plunger through its rod tube to the corresponding pair of pressure pumps under ground, and this actuates the working plunger connected with it, either drawing or forcing water, the other

pair acting conversely. The water is forced into an air vessel, and thence through the rising main in one lift to the surface, the power supplied by the descent of water in one column being nearly sufficient to effect its return in the other. The tubes were proved to 100 atmospheres. The working pressure on the underground pumps (due to the difference between their areas and those of the pumps at the surface) is 50 atmospheres, and the hydrostatic head in the rods is 27 atmospheres. The total working pressure, including friction, is 77 atmospheres, or about 1155 lbs. per square inch. The engine is worked at a speed of 10 double strokes per minute, the delivery of water being continuous. Careful observations were made in order to ascertain the work absorbed by the friction of the different parts of the machinery, and it was found to be from 25 to 29 per cent. of the total power developed. The effective work of the pumps, at 10 double strokes per minute, was 100 HP, and the indicated HP of the engine, with a mean pressure of 20 lbs. per square inch on the piston, was 136 HP, which gives a combined efficiency of 75 per cent.

Where small heads of water of 70 feet or so have to be utilised, and where the power is required continuously, a water-wheel or turbine possesses advantages over an hydraulic engine, which it is necessary to make large and cumbersome. If high speed is required, then a turbine might be used even with a greater head, as the speed involved by applying high pressure water to a turbine is not a disadvantage. The skin friction, which is so important a factor in turbines, is reduced to a minimum where they are small and work at high speeds. If a slow motion, on the other hand, is wanted, and a high pressure of water has to be utilised, the power had better be applied to a direct-acting pump. A large turbine working slowly under a great head would involve serious loss from skin friction in the turbine itself, as well as from friction in the gearing which would be required.

THE ACCUMULATOR.

Sir William Armstrong devised this simple means of obtaining pressure on a column of water by a weight instead of by elevation. The first accumulator (which is still in daily work at Elswick) has the ram attached to the ground, the cylinder rising and falling. The cylinder is encased with cast iron weights, which are held together by iron hoops. In the usual form of accumulator the ram rises instead of the cylinder, as is shown in vertical section by fig. 10. A cylinder A contains a ram B, upon the top of which is attached a crosshead B′, carrying a loaded case CC, which is suspended to the crosshead by a series of wrought iron straps bolted to it and also to the case. The weight placed in the case can be varied to suit the pressure required, and as the weight rests on the top of the ram it follows that whatever water is pumped into the cylinder from the engine through the pressure pipe D, will be subject to that pressure. A stop valve E enables the water to be cut off. When the ram has risen to the top of the stroke (and the cylinder is full of water under pressure), it stops the engine by means of a chain connecting with the steam throttle valve of the engine, and water ceases to be pumped into the accumulator. When the ram falls (owing to the abstraction of water from the cylinder), the steam throttle valve is opened, the engine works again, and water is pumped into the accumulator.

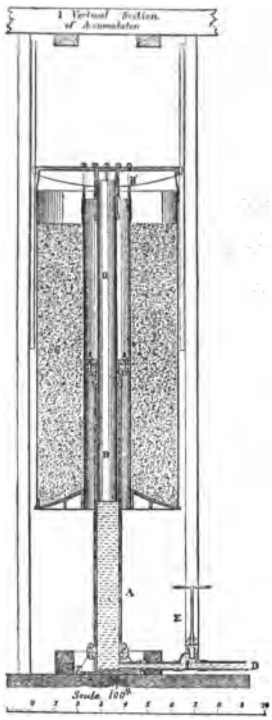
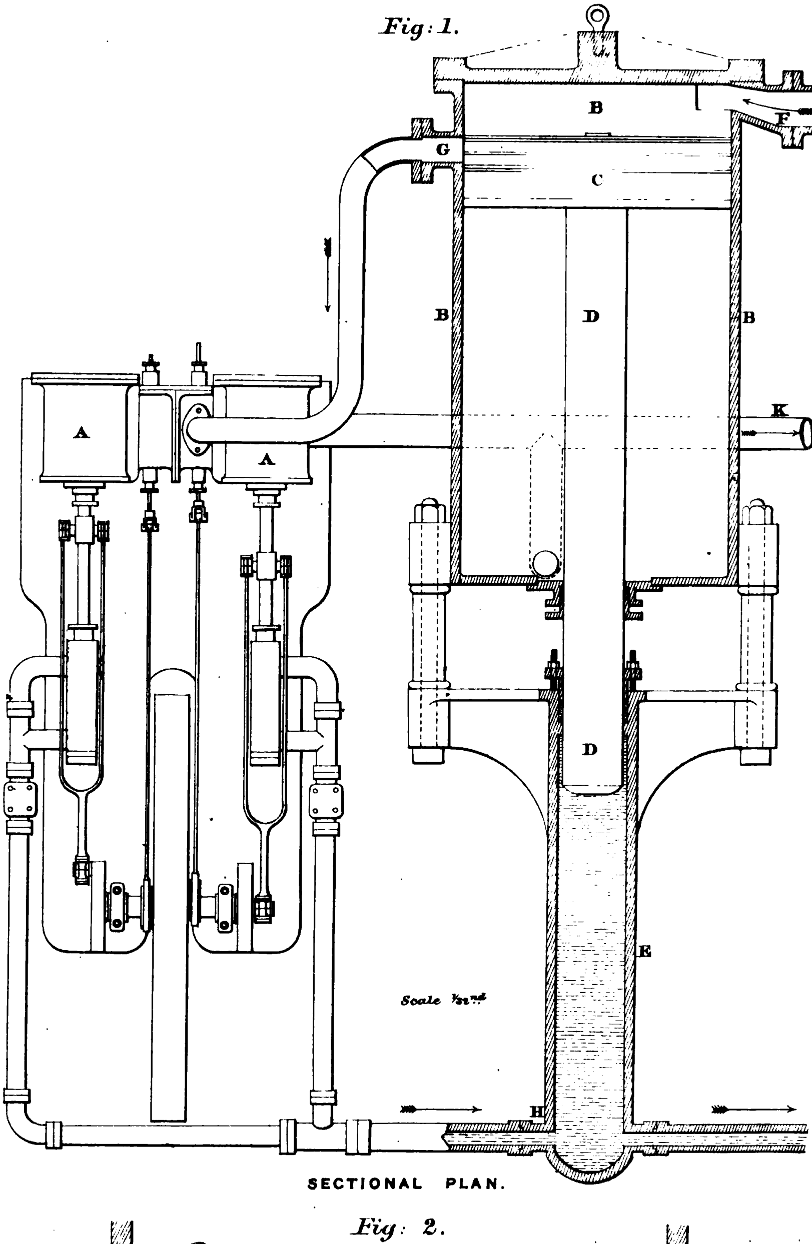


Fig. 10.

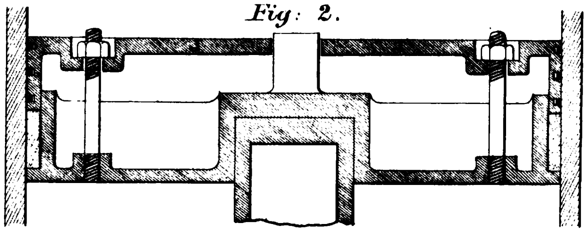
The largest accumulator that has yet been made is at Liverpool. It is 23 inches in diameter, and has a 40-feet stroke.

Fig: 1.



SECTIONAL PLAN.

Fig: 2.



SECTION OF PISTON.

Scale 1/16th

Thos Kell & Son, Lith.

By means of an accumulator an artificial head can be maintained at any part of an hydraulic main. The abstraction of the high pressure water to actuate hydraulic appliances is practically intermittent. The supply of high pressure water from the pumping-engine is continuous. It follows, therefore, that as the transmission of pressure through a water main is practically instantaneous, the intervals (however small) between the time of the production of the power and its utilisation in the appliance, enable the pressure to be maintained, and the excess to be stored in the accumulator. The variation in the consumption of the power, by reason of the fluctuation in the working of the machines, is at once adjusted by the accumulator, which both serves to store up the excess of power delivered to the mains from the engines, and also to maintain the pressure in the mains. By this means power is transmitted without practical loss in hydraulic mains. It has been found that water at 700 lbs. pressure can be transmitted a mile with a friction-loss of only about 2 per cent., where the mains are properly proportioned and where the head is maintained.

The amount of useful work which is stored up in an accumulator when the ram is at the top of the stroke is ascertained in the following way :—Taking a 12-inch accumulator having a stroke of 22 feet, and working at a pressure of 750 lbs. per square inch :—

Area of 12-inch ram = 113·097 square inches.
22-feet stroke = 264 inches.

$$\therefore \text{Power} = \frac{113{\cdot}097 \times 264 \times 750}{33{,}000} = 678{\cdot}582 \text{ HP.}$$

This store of power is capable of being given out as required, either quickly or slowly, according to the working of the appliances.

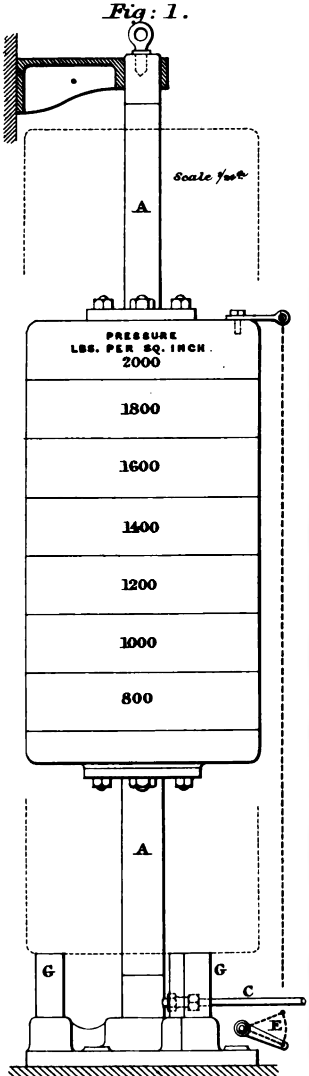
Mr. Andrew Betts Brown has devised an accumulator by applying steam to one side of a piston which acts upon a water ram. This is shown by Plate 2, which is taken from the *Proceedings of the Institution of Mechanical Engineers.* This accumulator consists of a large steam cylinder 36 inches in

diameter, fitted with a piston C, and a piston rod D, which forms the ram of a hydraulic cylinder E, having $\frac{1}{15}$th the area of the steam cylinder B. A steam pressure of 50 lbs. per square inch, therefore, gives a water pressure of 750 lbs. per square inch in the hydraulic cylinder (less the amount of friction). Steam is admitted to the top of the accumulator cylinder (at F) from the ordinary donkey boiler, or the main boilers. The pumping-engines AA are supplied by a branch, G, from the opposite side of the cylinder, and deliver the water from their pumps into the hydraulic cylinder at H. The bottom of the accumulator-cylinder B is open constantly to the exhaust K. When steam is turned on to the accumulator, the engines start, at the same time pumping up the hydraulic ram D, and they continue working until the steam-piston rises high enough to close the steam-pipe orifice G. The engines then stop, but when water is drawn from the accumulator by the action of the hydraulic machinery, the steam-piston descends, maintaining the pressure of 750 lbs. per square inch upon the water; at the same time, by opening the steam-pipe G, it starts the engines again, by which the accumulator is replenished.

An accumulator has been designed by Mr. Tweddell to meet the variation of demand for high pressure water, such as arises when only one appliance is at work in a system of hydraulic pipes supplying numerous shop tools. This is shown by Plate 3, from the *Proceedings of the Institution of Mechanical Engineers.* The ram or spindle A of this accumulator is fixed, and acts as a guide, whilst the cylinder B slides upon it, and is loaded with the weight necessary for giving the required pressure to the water. The water is pumped in at the bottom at C, and fills up the annular space surrounding the spindle. The whole weight has to be lifted by the water acting only on the shoulder D, which is made by a brass bush $\frac{1}{2}$ inch thick all round the spindle. A compact arrangement is thus gained, and any required cubic capacity is obtainable by lengthening the stroke. The accumulator is supplied by two pumps, each $1\frac{3}{8}$ inch diameter and $3\frac{1}{2}$ inches stroke, running at about 100 to

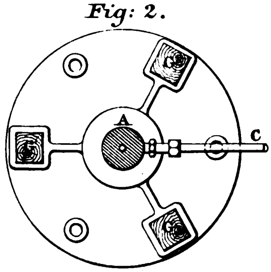
ACCUMULATOR. PLATE. 3.

Fig: 1.



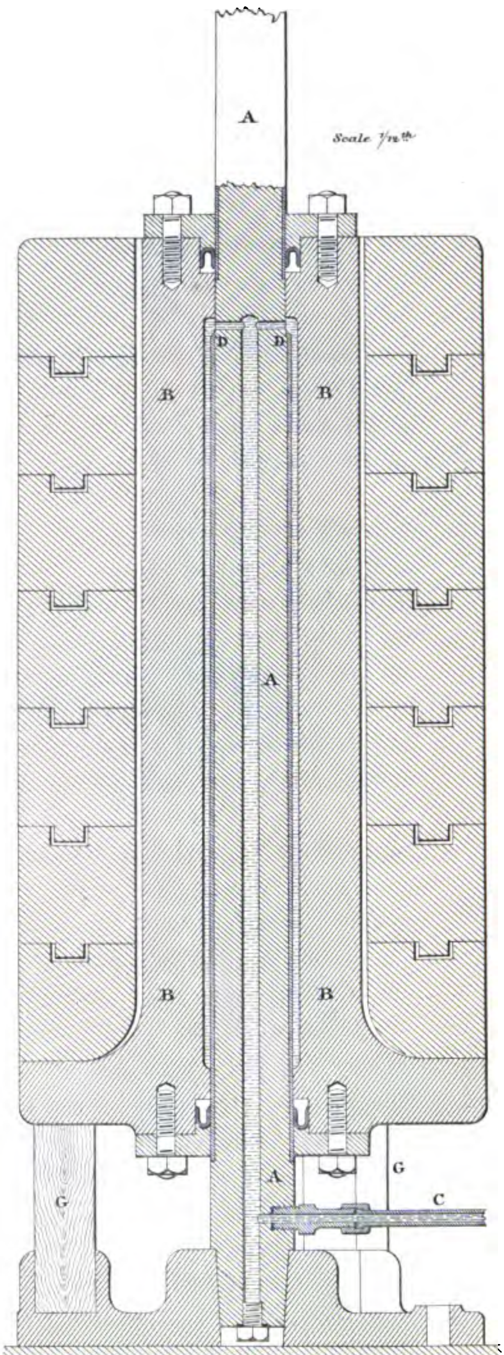
ELEVATION.

Fig: 2.



PLAN AT BOTTOM.

Fig: 3.



VERTICAL SECTION.

Thos Kell & Son, Lith.

120 revolutions per minute. When the loaded cylinder B reaches the top of its stroke, it is made to close the suction cock E of the pumps, thus stopping the supply of water. When it is desired to put in a new packing-leather at the bottom, the weighted cylinder is let down to rest upon blocks placed on the wood chocks G at the bottom, and the spindle is drawn up out of its tapered seat by the eye-bolt at the top. To renew the top leather, the bracket holding the top end of the spindle has to be removed. This accumulator (having only a small area) falls quickly when the water is withdrawn, thus producing a combined blow and squeeze, which is of great advantage in hydraulic rivetting.

What is known as an "Intensifier" was made at Elswick many years ago (for the Russian Government) to test gun barrels before they were put together. There were three rams, the two outside for lifting the load, and the centre (a steel ram of small diameter) on which the load was allowed to rest on the down-stroke. The two outside rams lifted the load up, and when it was at the top the pressure was taken from the outside rams, and the load was allowed to fall. The rams and load were proportioned to give a test of 7 tons to the square inch.

A means of intensifying pressure is shown by the accumulator at fig. 11 (from the *Proceedings of the Institution of Mechanical Engineers*). A pipe A conveys low pressure water into the cylinder B. The pressure on the piston C acts upon the smaller ram D, and gives an increased pressure to the water in the second cylinder E, in proportion to the relative areas of the piston C and the ram D. In the illustration the piston is 19 inches, and the ram $3\frac{3}{4}$ inches, in diameter. A pressure of 60 lbs. per square inch on the piston gives 1540 lbs. per square inch on the ram. The water from the pumps enters through the inlet G, and passes out at H to the machine to be worked by it. No water is consumed from the low-pressure cylinder, but it is simply driven back by the force pumps into the low-pressure accumulator or mains.

The loss of useful effect between the pumps and a properly

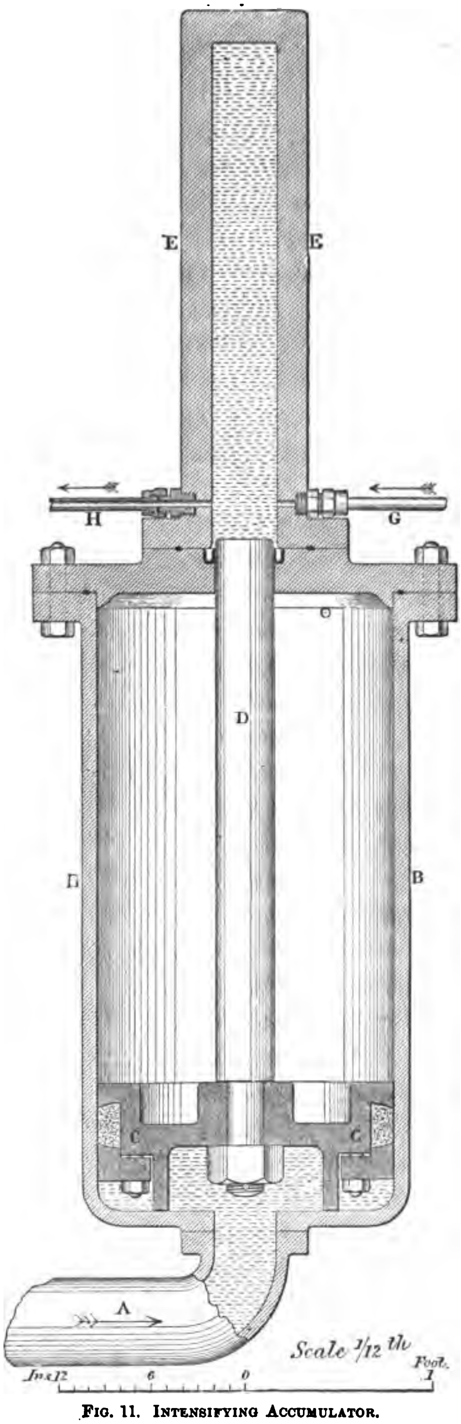


Fig. 11. Intensifying Accumulator.

packed accumulator is but trifling. Experiments carefully made by Mr. Tweddell are recorded of the working of two pumps delivering water into an accumulator. The exact height that the ram was raised by the strokes of the pump was registered. With one pump working, 1694 cubic inches was the theoretical delivery of the pump for 20 strokes, and 1614 cubic inches was the actual quantity pumped into the accumulator, showing a loss of only 4¾ per cent. With both pumps working, the corresponding quantities (for 20 strokes) were 3388 cubic inches, and 3278 cubic inches, showing a loss of only 3¼ per cent. It was noticed in these experiments that 1250 lbs. per square inch was the ascending pressure in the accumulator, and 1225 lbs. per square inch was the descending pressure. In ascending, the friction had to be overcome by the pump in addition to lifting the load, and in descending, the friction had to be overcome by the load itself; the amount of the friction will therefore be half the difference of pressure in the two cases, or 12½ lbs. per square inch, which is equivalent to 1 per cent. of the power.

Messrs. Clark & Standfield have arranged a differential accumulator for working hydraulic lifting-presses. In their accumulator the dead-weight of the machinery, which has to be raised and lowered, is constantly balanced, so that only a small additional power is required to give it motion. Three accumulators, rams, or plungers, are usually employed, two of which are of such dimensions as to produce a pressure on the ram of the hydraulic press exactly sufficient to balance the dead-weight of the machinery carried by it. In this condition the accumulator and the hydraulic press are in equilibrium, and a very small increase or decrease of pressure suffices to cause the hydraulic ram to ascend or descend, as the case may be. An extra load is then put on the accumulator sufficient to cause it to descend, and to raise the hydraulic ram with any desired load upon it. The third plunger is placed centrally under the head of the accumulator, and the pipes communicate with the two outer plungers so that all three can be connected at will. When the accumulator is down, and the ram is elevated (as just

described), then if the communication is opened between the three plungers, the weight of the accumulator, which was at first supported by two plungers, being now supported by three, the pressure on the water is diminished, and consequently the accumulator ascends and descends. In order to raise it again it is only necessary to allow the water to escape from the central ram, when the whole weight becomes supported on the two plungers as before, and the pressure is consequently increased and the ram again ascends. A small pump is employed to keep the accumulator charged.

In order still further to diminish the loss of power entailed by hydraulic rams when raising and lowering heavy weights, Messrs. Clark & Standfield compensate for the varying immersion of the ram. When a ram is raised in the ordinary manner it is evident that as it ascends out of the water into the air, it increases in weight, and its balancing power diminishes by an amount which is equal to the weight of a column of water of its own bulk. Similarly, as the plunger of an accumulator descends, it loses a weight equal to the bulk of the column of water which it displaces, and both of these actions concur to diminish the power of the machine more and more as it approaches the full extent of its stroke. To obviate this, the load on the accumulator is increased, as its plunger descends, by a weight of water sufficient to compensate for the varying immersion of the plunger and of the ram of the press. By this means the dead-weight of the machine itself is counterpoised in every position, and the only power required to work the machine is that which is requisite to raise the load itself and to overcome friction. By the same means increased power is given at the end of the stroke, by adding to the load a greater weight of water than is required for compensating for the varying immersion of the rams and plungers. Conversely, decreased power may be given at the end of the stroke by causing weighted tanks or vessels (which form the load of the accumulator) to descend into water.

Where it is desirable that two or more rams should ascend

synchronously through equal distances (as, for instance, in the two ends of a bridge or canal lift, or in raising guns) two or more plungers are combined into a group beneath one accumulator, so that as the plungers descend, all the rams ascend through uniform distances. In order to cause all four corners of a bridge or other moving apparatus (which is supported by presses at its two ends) to ascend or descend in a horizontal position, means are provided for allowing an escape of water from beneath either of the rams, if from any cause one of them should become elevated above the other.

An accumulator on the compensation principle is designed by Messrs. Clark & Standfield for raising and lowering a gun. It is constructed with three presses and rams, the central one being smaller than the outer ones. These are loaded with a weight so adjusted as to balance the gun, which is supported on a ram working in a separate press. The three plungers of the accumulator-presses are of such dimensions that when they are loaded with the weight, and connected jointly with the ram of the gun-press, they just balance the weight of the gun, which is therefore free to be raised or lowered without any power except that which is necessary to overcome friction. When this equilibrium has been obtained, a small additional weight is added on the accumulator, which consequently descends and elevates the gun to its full height. All this time the small centre ram of the three is out of action, and is merely connected by a pipe with the supply reservoir. If it be desired to cause the gun to descend, a tap in connection with the small cylinder is opened, so as to place the three plungers in communication. The pressure being now distributed over all three plungers, instead of over only two, the weight ascends, and the gun descends. If it be again required to raise the gun, it is only necessary to close the tap, and the weight of the accumulator, being only on the two plungers, again causes the gun to ascend. In this way the gun may be raised and lowered at pleasure by the mere turning of the tap; the only power that is wasted is that of the small plunger, which is made of such size as to be

just sufficient to overcome friction. The same effect of obtaining a slight variation of pressure in the accumulator may be produced by either allowing the before-mentioned small additional weight to rest on the larger permanent weight, or by holding it off therefrom. Thus if water under pressure is introduced below the central plunger, and the small additional weight is thereby raised, the gun will descend, but if this weight is allowed to rest upon the permanent weight, the accumulator will descend, and the gun will again rise.

The central press may be worked from any independent source of pressure without having a communication with the outer presses. In this case the weight is made heavier than the load to be moved, and in order to cause the load to descend, pressure is applied to the centre ram, and thus the load may be made to rise and fall at will.

If instead of raising and lowering a fixed weight, it be desired to raise a load, such as a waggon of earth, it is only necessary to increase the small additional weight to a sufficient extent, and to increase the size of the centre ram on which it rests; the ram will then ascend with its load, and when the waggon is deposited at the top, it can be caused to descend by opening the tap in the same manner as before.

To minimise the loss of power which is entailed in raising and lowering heavy weights by hydraulic rams, Messrs. Clark and Standfield provide compensation for the varying immersion of the rams and plungers at all parts of their stroke, by increasing the load on the accumulator as its rams descend. A tank of water is fixed above the accumulator, and a compensating ram works through a leather collar in the bottom of the tank, and rests upon the top of the accumulator. The upper portion of the compensating ram is always under water, and its sectional area is equal to twice the sectional area of the three rams of the accumulator. When the plungers are at the bottom of their stroke, the compensating ram is at the bottom of the tank, and supports the whole weight of the column of water above it, but when the plungers are at the top of their stroke, the

top of the compensating plunger is near the surface of the water, and consequently supports no pressure except that of its own weight.

If the proportions above-named be carried out, it follows that whatever be the number or dimensions of the plungers or rams, they will always be compensated for at every portion of the stroke. Compensation is made for the rise and fall of the water in the tank (caused by the rise and fall of the plunger in it) by making the compensating ram a little larger than the proportions before stated, in the ratio which its sectional area bears to that of the tank.

Many years ago, Sir William Armstrong arranged for Mr. Ure, the engineer to the Tyne Commissioners, an air accumulator to work at about 250 lbs. to the square inch. He had some hydraulic cranes put on board a screw hopper barge, used for discharging ballast from vessels lying in the pool of the river, and taking it for deposition out to sea. These cranes lifted 2 tons, and were able to discharge 60 tons an hour. They had hydraulic lifting, turning, and traversing motion applied to them. As it was not considered practicable to introduce an accumulator on this small barge, a cast-iron air vessel was adopted to work at 1000 lbs. on the square inch. Some difficulty was experienced in the first instance in working the air pump with this high pressure, but by introducing a small stream of water with the air on the suction side, and by allowing the water to fill up the spaces between the ram and the valve, not only was all difficulty overcome, but even a higher air pressure was able to be used. Before this small stream of water was employed, a certain amount of air remained in the cylinder, and as this air was not forced out by the plunger, it prevented the full amount of air at the end of the stroke from being sucked into the cylinder for the next stroke. The introduction of a small quantity of water caused this space to be taken up by the water, and so that difficulty was overcome. Owing to the air getting mixed with the water, a cream was formed, which was obviated by the application of water in the small air-pump. These cranes are still at work.

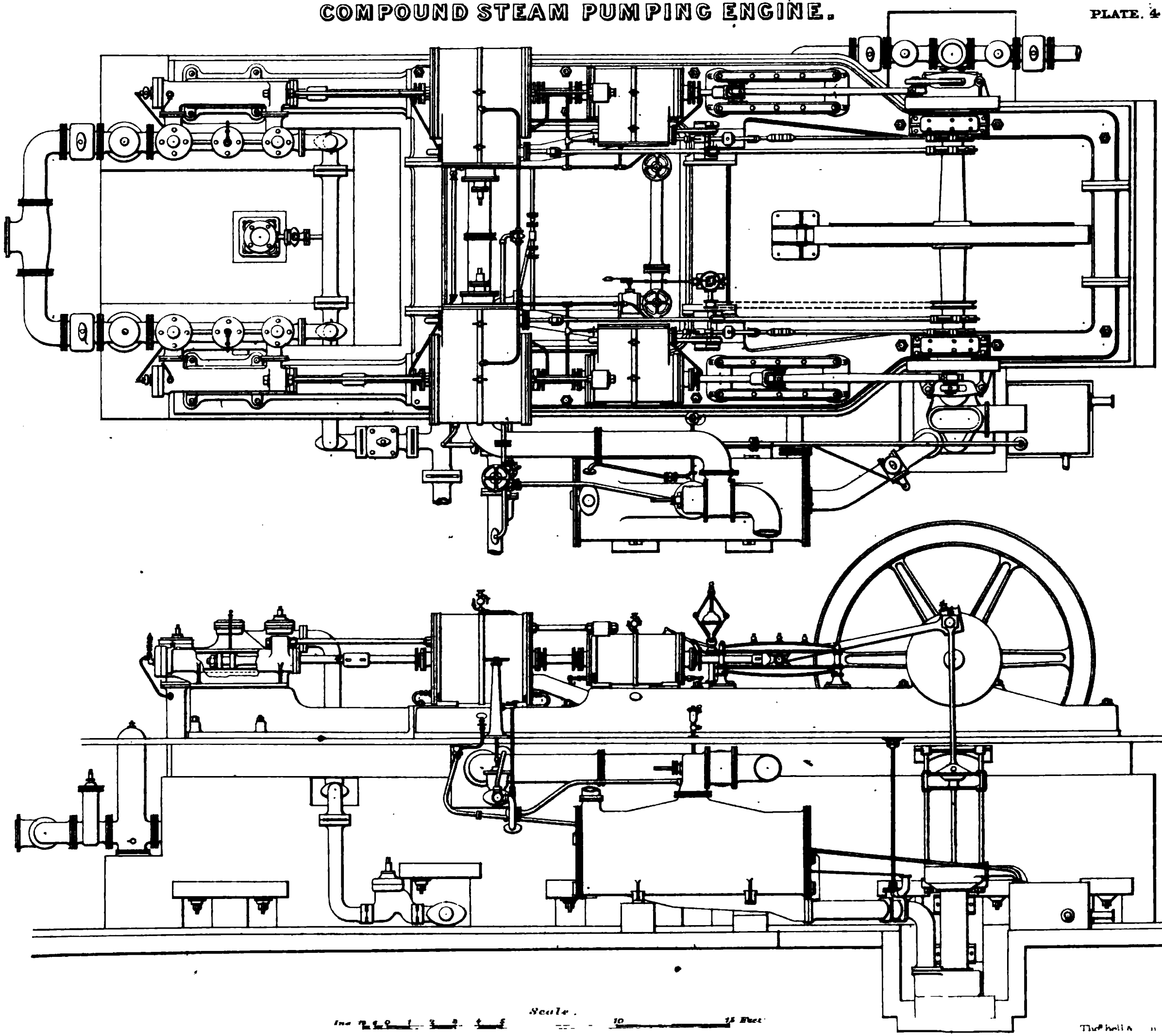
On board one of the P. and O. steamers (the *Massilia*) hydraulic machinery is now employed in which air vessels are introduced, working at a pressure of 1000 to 1200 lbs. on the square inch. These air vessels are composed of wrought iron pipes 8 inches in diameter.

HYDRAULIC PUMPING-ENGINE.

The first hydraulic pumping-engines were high pressure, with plungers at each end of the pistons, and in a direct line with them. The original valves were made in the first place with flat faces of cast iron, but these were very soon cut. Faces of steel were then tried, but they also did not stand for any length of time, and eventually mitre-valves of hard gun-metal composition were introduced, which have since become universally adopted. A broad flat surface was first employed, with the idea that in working at high pressure the beat of the valve on the surface would require a separate face. The effect, however, of this was that the water passing between the broad flat areas scarred the metal, and in addition, the annular surface under the accumulator required a greater pressure on the plunger to lift the valve. The consequence was that the metal of the pump became strained and caused a loud report, which was due to the expansion and subsequent contraction of the metal. This was at first regarded as a beat of the valve, but subsequently it was observed that during a short interval the pressure on the pump-barrel ran up very much above the accumulator pressure, owing to the latter acting on the large annular surface of the valve. On one occasion when the engine was worked rapidly, the violence of the shock burst the pump-barrel, although it was exceptionally strong. This concussive action was made the subject of a communication to the Institution of Civil Engineers by Sir William Armstrong as far back as 1853. He pointed out that, in all cases in which pumps are to be worked rapidly against a heavy pressure, it is important that the area of the valve which is acted

COMPOUND STEAM PUMPING ENGINE.

PLATE. 4.



Scale.

upon from beneath should bear a large proportion to the area that is pressed upon from above. As a general rule, concussion arises on the fall of the valve, and is caused by the valve having an excessive rise which involves too much time for closing, so that there is a slight interval on the turn of the stroke when the valve is open, and is then suddenly forced down by the pressure of the return stroke. Another cause of concussion is the momentum imparted to the delivery water by the previous stroke of the pump producing an overrunning of the column in the delivery-pipe. To provide against an excessive rise of the valve it is usual to make them of large dimensions, and with two or more bearing-faces, having a small rise for each; at the same time a large area is provided for the passage of the water.

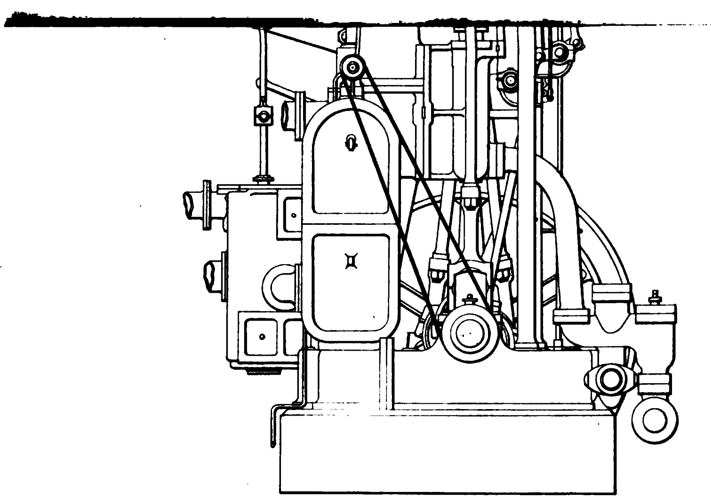
One of the most recent arrangements of engines for pumping high pressure water is that shown by Plate 4. This is an Elswick high pressure, compound, condensing steam pumping-engine. It is on the double tandem type, having two high and two low pressure steam cylinders arranged one behind the other. The pressure pumps are in the rear of the low pressure cylinders, and are worked direct by a prolongation of the piston rod, which is common to both cylinders. There is a cylindrical surface-condenser with air-pump placed vertically, and worked directly off one of the crank-pins. The circulating pump is similarly placed and is worked off the other crank-pin. The engine is arranged to work without the condenser when required, provision being made for exhausting into the air. The high pressure cylinder has double slides, the point of cut-off of the expansion slide being varied from the outside of the steam chest. The pressure pumps have mitre valves. The rams and pistons are single-acting in suction, and double-acting in delivery. The engine is self-contained, and the bed-plate takes all strains, only needing to be bolted down to a simple masonry foundation. A governor for preventing racing is provided, and also a spring-loaded plunger on the delivery main, to absorb shocks of the water column.

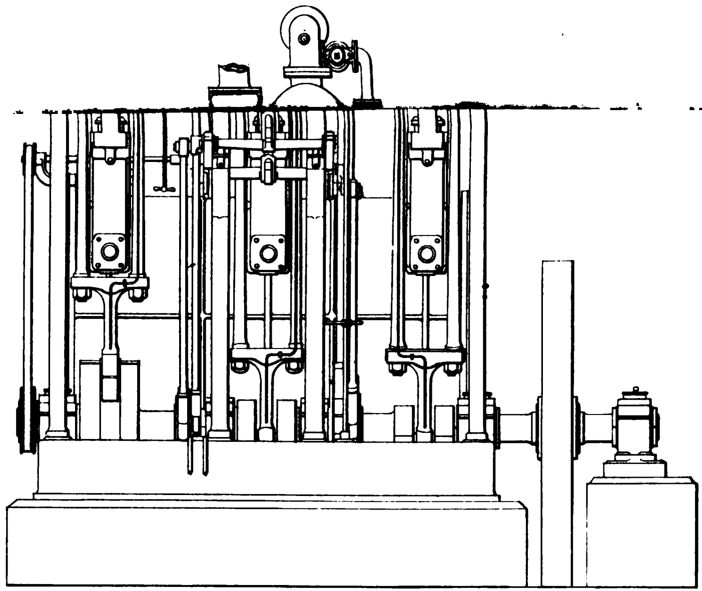
The Hydraulic Engineering Company of Chester have adopted Three-throw Compound Engines at the London Hydraulic Power Company's central pumping station. The centre cylinder is high pressure 19 inches diameter, 2-feet stroke. The two outer cylinders are low pressure, 25 inches diameter. The pump plungers are 5 inches diameter, and the quantity of water delivered into the accumulators under trial was 296 gallons per minute. The work stipulated for was 140,000 gallons in ten hours, the actual quantity pumped being 156,480 gallons, or an average of 260 gallons per minute. The engines are fitted with surface condensers, and with air and circulating pumps. They will stop and start in any position, make a single revolution, and stop again. The indicated HP gave a maximum of 205 HP, and 84 per cent. of efficiency in water pumped. The consumption of small coal in Lancashire boilers, with Vicars' stokers, was 2·4 lbs. per indicated HP, and the weight of steam 20·7 lbs. per indicated HP. The main valves are made on the balanced type of lift valve, with a waterway the full area of the pipe. The balancing is effected by a small controlling valve 1½ inch in diameter, inserted inside the large valve, by which only the small valve and the weight of the large valve have to be raised. A man can, with a 12-inch lever, lift and lower a 6-inch valve, under a pressure of 700 lbs., which is equivalent to a load of 8·8 tons pressing the valve on its seat. A compound hydraulic engine made by the Chester Company is shown on Plate 5.

In ordinary forms of engines for pumping high pressure water, the piston speed is about 300 feet per minute, although the engines can be run at 400 feet, or even more, if occasion require.

The economical production of hydraulic power is essentially necessary to be kept in mind. In this connection the type of engine employed is of paramount importance. The example given of the Elswick compound condensing engine could be multiplied by others of well-known makers, such, for instance, as the "Cowper" engine made by Messrs. Simpson & Co., of

COMPOUND HYDRAULIC ENGINE.





Thos Kell & Son, Lith.

London. In this design of engine two cylinders are placed side by side, one being high pressure and the other low pressure. The steam on its way from the high to the low pressure cylinder passes through a steam-jacketed receiver with an internal liner, which ensures the steam being dry before it goes into the low pressure cylinder. From trials of several engines made by Messrs. Simpson & Co., the following data are deduced, which show the small amount of coal that is consumed, and consequently the economical production of power which is now attained:—

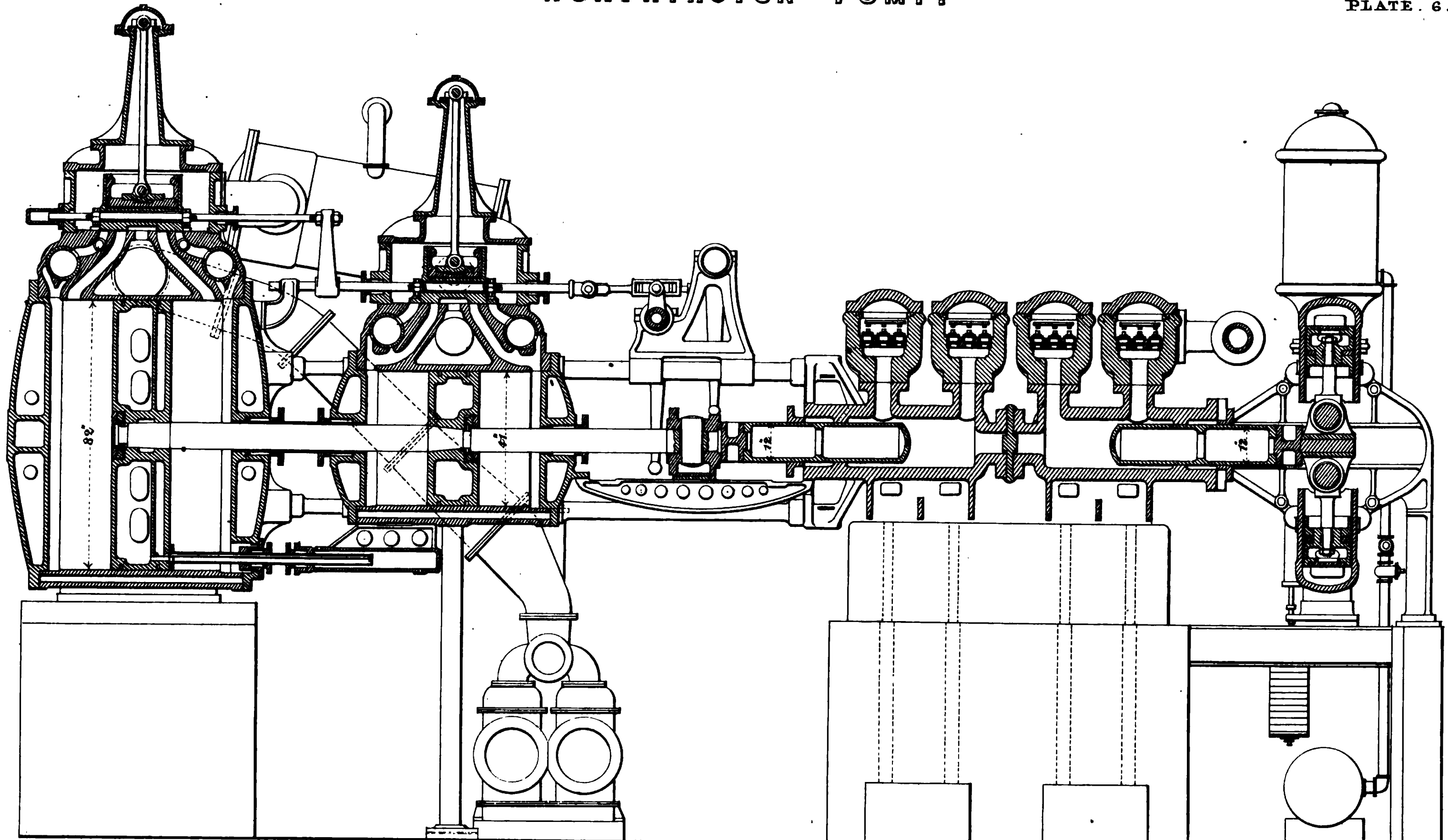
	Steam per 1 HP per hour.	Coal per 1 HP per hour.
Cornish bull engine	32·2	3·17
Cornish beam engine . . .	24·15	2·44
Single cylinder engine . . .	21·3	2·15
Woolf compound engine . . .	15 4	1·6
Cowper compound engine . . .	14·84	1·53

Where the pumping-engines are of the crank and flywheel type, the pump-pistons move at a variable speed according to the angularity of the connecting-rod, and the quantity of water that is delivered varies at each instant, from zero at the ends of the stroke, to a maximum about half stroke, when the pistons are moving with the same velocity as the crank-pin. This variable delivery produces a change of velocity in the rising main, and where the engine is pumping through a long main, or one which contains a large body of water, very severe pressures are caused in the pump by the changes of velocity and the inertia of the water. This variation of pressure is compensated for by air vessels, otherwise the pressures set up in the pump are sufficiently great to fracture the rising main or pump-work. In cases where (through the heavy pressures in the air vessels) difficulty exists in retaining the air, advantage is experienced in adopting a pump in which the delivery of water is constant, and is not controlled by a crank and flywheel. The "Worthington" form of pump is attracting much attention

now, as it fulfils the conditions referred to, the delivery being uniform at all parts of the stroke. There are two pumps, each double acting, the flow from one dovetailing into the flow from the other. The steam cylinders are directly in line with the pumps, and there are no cranks or flywheels. This system has been adopted in pumping oil in America, where, owing to the great length of the mains and their smallness, the head on the pumps is all due to friction. When oil was forced with pumps whose motions were controlled with cranks, such excessive pressures were set up, owing to the change of velocity in the mains, and consequent increase of frictional head, that the pipes were continually bursting.

Plate 6 shows an engine made in America, which for size and power surpasses anything that has been constructed in this country. It is about 800 HP. The low-pressure cylinders are 82 inches, the high-pressure cylinders are 41 inches, and the pump plungers are 12 inches in diameter. The pressure they force against is 1500 lbs. per square inch, which is equivalent to a dead pressure of 151 tons, nevertheless the engine works perfectly and noiselessly. The steam pressure is 100 lbs. per square inch, worked at a high rate of expansion. The combination of an uniform pressure of steam in the engines with a high rate of expansion has been a matter of much comment, and is explained by Messrs. Simpson (the makers) in the following way. With compound pumps, as formerly constructed, steam was admitted to the small high-pressure cylinder at its full pressure for the whole length of the stroke, and was then exhausted into the large low-pressure cylinder to do duty on the return stroke at a lower pressure, but on a very much larger area of piston. By this means, with an initial pressure of about 80 lbs., a mean effective pressure was obtained for the two cylinders working in tandem, that would be equal to about 40 lbs. on the large cylinder with pumps of the usual proportions. With the high-duty engine, the object is to get a result equal to any degrees of expansion up to the possible economic limit, and at the same time to preserve all the approved features

WORTHINGTON PUMP.



Thos Kell & Son, Lith.

of the pump as originally constructed. The apparatus to accomplish this consists of two opposed small oscillating cylinders connected to an extension of the plunger-rod of each set of cylinders, as shown in the cut at the water end. These cylinders and their connections are filled with water (or other liquid). Compressed air from an accumulator or storage reservoir is admitted to the surface of the water (or other liquid) that goes into these small cylinders, at a pressure suitable to the duty to be accomplished, for the purpose of maintaining a constant load at a practically constant pressure on their pistons through the medium of the interposed liquid.

These pistons act in opposition to the engine up to half the stroke, during which time the steam in the high-pressure cylinder may be at its initial pressure, then the point of cut-off may be established, and, as the steam-pressure diminishes, the force that is stored by the compression is given off and is restored to the source from which it came, securing a practically constant exertion on the piston-rods and water-plungers throughout their whole stroke.

The two small cylinders for the reciprocation of power are placed directly opposite to, and balance, each other, thus relieving their cross-head from any side strain on the slides.

For slide-valve engines of this description the cut-off can be fixed at a suitable fraction of the stroke of the small cylinder, or, in other words, the steam may be taken at a high pressure until it only partly fills the small cylinder, then it is expanded to the end of the stroke and admitted to the large cylinder to be further expanded.

Where the most economic results are desired, the low-pressure as well as the high-pressure cylinders on the engine are provided with cut-off valves. These consist of semi-rotating plug-valves placed in the admission ports of the cylinders, and are worked by means of the direct connections shown in the Plate. The point of cut-off can be fixed by experiment for both cylinders, and need never be altered while the duty remains the same.

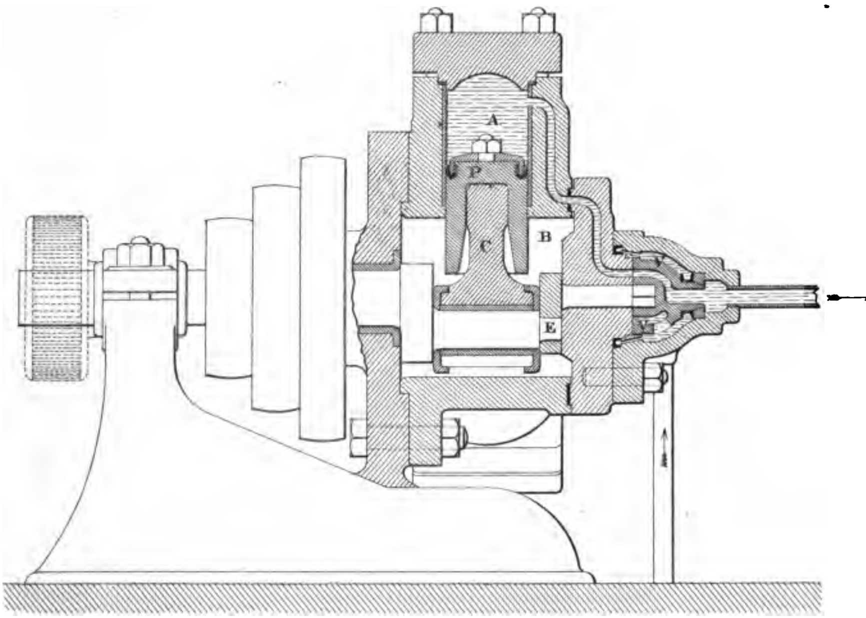
One of these engines was carefully experimented on by Mr. Mair (of Messrs. Simpson & Co.) in America, and he found that as high a degree of economy was obtained as with any type of crank and flywheel engine, whilst a perfectly uniform stream of water was delivered.

THREE-CYLINDER ENGINES AND CAPSTANS.

For some time after the introduction of hydraulic machinery for purposes to which a reciprocating motion was required, great difficulty was found in adapting it to obtain rotary motion, and many forms of engines have been invented for that purpose. On the continent Haag's engine has been largely employed; in this the oscillation of a cylinder on its axis alternately opens and closes the ends of the axis to admit and exhaust the water. Ramsbottom devised an engine with three cylinders oscillating on a cast-iron pipe, which was divided by a longitudinal diaphragm into two sections, one of which supplied the water to the cylinders, and the other exhausted the water from the cylinders through ports. The Brotherhood three-cylinder engine (made by the Hydraulic Engineering Company of Chester) is an excellent appliance for producing rotary motion by water-pressure. A description of this was given to the Institution of Mechanical Engineers, and is shown by Plate 7, figs. 1, 2, and 3. Three cylinders A (made in one casting) are always open at their inner ends, and are attached to a central chamber B. They contain three pistons P, which transmit motion to the crank-pin through the struts C. The water is admitted and exhausted by means of the circular disc valve V, with a lignum vitæ seating. The valve is rotated by the eccentric pin E. A face view of this valve is shown in fig. 3. It has segmental ports which, in rotating, pass over apertures in the valve seating. There being no dead centre, the engine will start in any position of the crank-pin, and a perfectly uniform motion of the shaft is obtained without a flywheel.

BROTHERHOOD'S THREE CYLINDER HYDRAULIC ENGINE.

Fig: 1.



LONGITUDINAL SECTION.

Fig: 2.

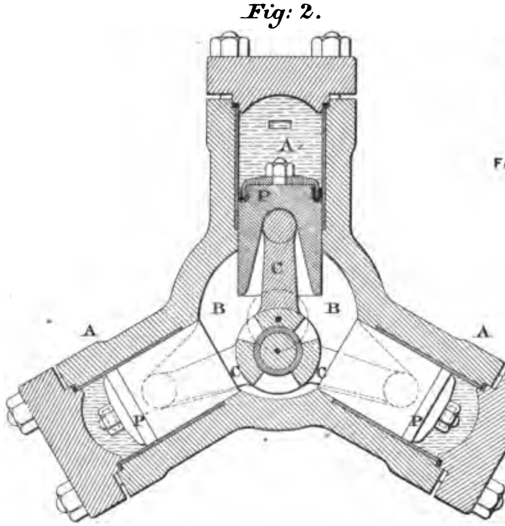


Fig: 3.

FACE OF VALVE.

TRANSVERSE SECTION.

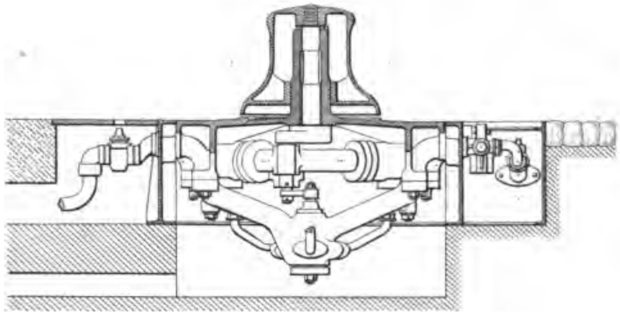
Scale 1/12th

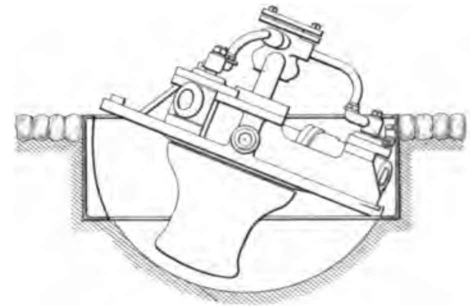
Inches 12 6 0 1 2 *Feet*

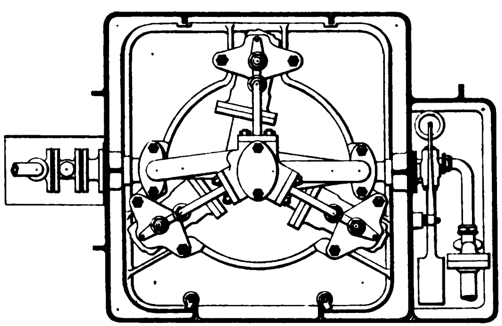
Thos Kell & Son, Lith.

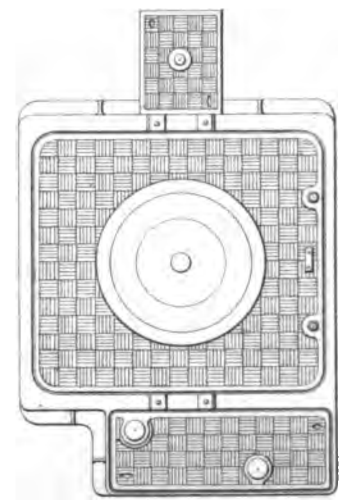
TURNOVER HYDRAULIC CAPSTAN.

FOR HAULING RAILWAY WAGONS.









Scale.

Inches 12 9 6 3 0 1 2 3 4 5 Feet

Thos Kell & Son, Lith

The pressure is always on the outer end of the piston, so that the struts C′ are in compression, and take up their own wear. This engine is well adapted for transmitting pressure to appliances which are worked intermittently, as, owing to the great speed at which it can be run, it will not only save the loss from friction (where gearing is employed), but will also reduce the friction in the machine itself, by enabling the gearing for increasing speed to be dispensed with.

The production of a simple hydraulic rotary engine led to its application to capstans, several forms of which have been introduced. One of the most recent and compact is that made at Elswick, which may be termed the Turnover Hydraulic Capstan, as shown by Plate 8. A bed-plate is hinged upon two trunnions, one admitting the pressure, and the other being used for the outflow of the exhaust water, after it has passed through the working valve or valves. To the bed-plate is cast a pillar, through which the crank shaft is guided, and to the other end of which the capstan head is fixed. To the single crank, on which the three rams act, is attached a cross rod communicating motion to a rotary valve, from which branch pipes convey the water to and from each cylinder. The trunnions of the bed-plate are carried on bearings attached to a cast-iron casing, which forms a framework for the capstan, and on which also is carried the working valve for regulating the starting and stopping of the capstan. This working valve is usually a mitred valve, to which a counter-weight and lever are attached, in such a manner that the counter-weight, when free, keeps the valve closed. To start the engines, the lever is pressed with the foot, thus raising the weight which keeps the valves closed. When the foot is removed the valve is closed, and the action of the capstan is stopped. The capstan is so balanced on the trunnions that it can be easily turned over by one man. The advantages of this arrangement chiefly consist in the facilities that are afforded for examining and oiling the parts. The capstan can be worked in any position, so that its action can be readily seen and adjusted. The usual power given to working

capstans is equal to a hauling power of about one ton on the rope, but smaller capstans than these are used where only one or two waggons are required to be moved at a time. The speed of the capstan can be varied from 2 or 3 revolutions per minute to upwards of 100 revolutions.

Mr. B. Walker has devised a double-acting capstan engine with two cranks set at right angles to each other. The arrangement of cylinders consists of two pairs, each pair being composed of a small and large cylinder, having a ram common to both, but of different diameters. The smaller rams are always in communication with the accumulator, whilst to the larger rams the water-pressure is admitted alternately by means of a slide valve, to which motion is given by the ram crossheads.

The introduction of hydraulic capstans into railway goods yards and other similar places, has proved of great advantage in expediting the operations of shifting trucks, making-up trains, and the like.

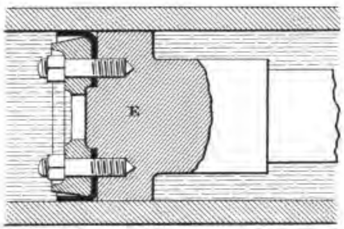
At the Toulon Dockyard, hydraulic capstans (with Brotherhood's three-cylinder engines) are worked at a pressure of 1500 lbs. per square inch. They are employed for hauling and lifting materials on the ships and ironclads. M. Berrier Fontaine is applying at this dockyard small Brotherhood rotary engines of ½ HP and 1 HP to drive Stone's flexible drills for drilling holes in the ships.

The efficiency of a three-cylinder hydraulic engine has been found to be over 70 per cent., when applied to a capstan working up to its full power.

MOTORS WITH VARIABLE POWER.

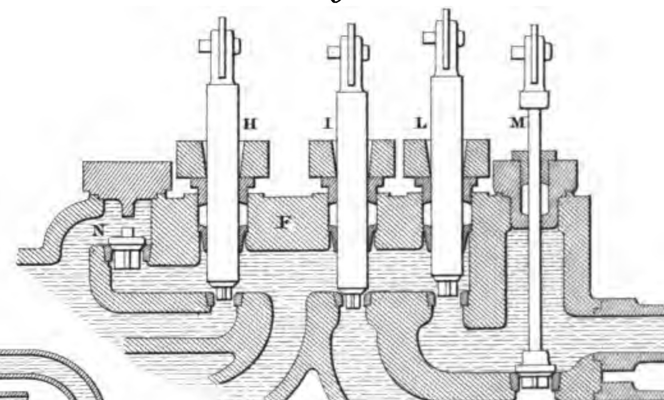
In utilising water-pressure, uneconomical results are produced in cases where the work to be done is variable, whilst the water that is consumed by the appliance is invariable. This was recognised by Sir William Armstrong at the commencement of his labours in the application of hydraulic power, and he met

Fig: 2.



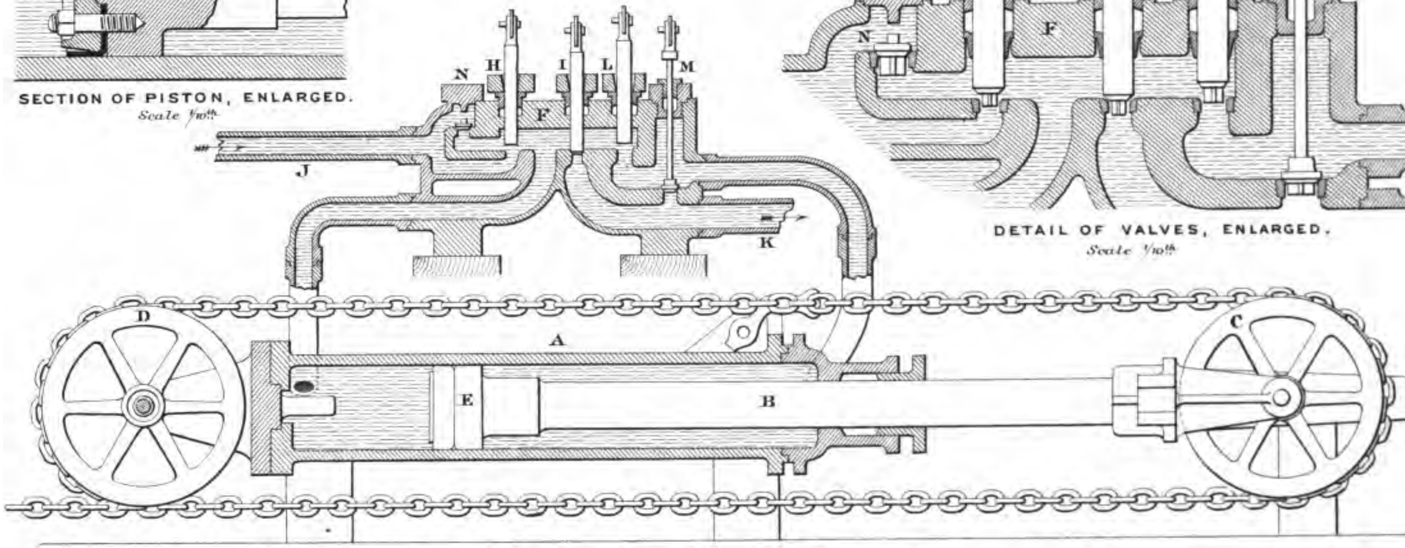
SECTION OF PISTON, ENLARGED.
Scale 1/10th

Fig: 3.



DETAIL OF VALVES, ENLARGED.
Scale 1/10th

Fig: 1.



DOUBLE POWER HYDRAULIC CRANE.
LONGITUDINAL SECTION OF CYLINDER AND VALVES.

Scale 1/20th

Ins. 12 6 0 1 2 3 4 5 Feet

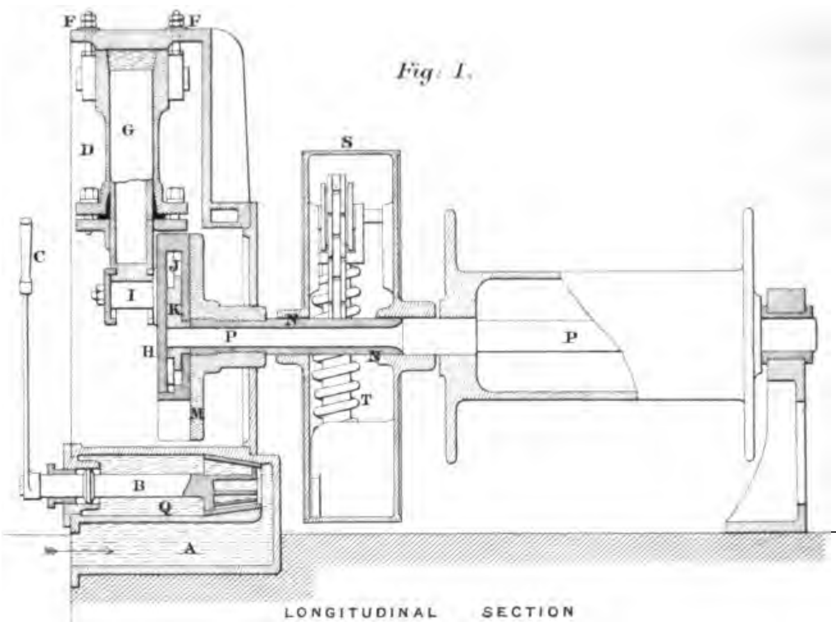
Thos Kell & Son, Lith.

the difficulty by employing what is known as a "double power," *i.e.* by using a combined piston and ram, as shown on Plate 9. Fig. 1 shows the arrangement of cylinder, piston, sheaves, chains, valves, &c., for a double-powered hydraulic crane. Fig. 2 is an enlarged section of the piston, and fig. 3 gives the details of the valves. A is the hydraulic cylinder, B the ram, and E the piston. The water from the accumulator enters the valve chest F through the pipe J, and the inlet valve H. By opening the valve L, water is admitted to both sides of the piston, so that the power that is exerted, and the water that is consumed, are due to the annular ring, or the difference between the areas of the ram and piston. In the event of the higher power being required, the valve L is closed, and the valve M is opened, whereupon, whilst the full area of the piston receives the pressure, the other side (or annular ring) is open to the exhaust K. For lowering a load the valves H and M are closed, and the outlet valve I is opened, which allows the water to escape from the cylinder into the exhaust pipe K. The valve L is also opened to allow the water to follow up the piston in the inward stroke.

Mr. Michael Scott has suggested the using of the waste water from cranes, by returning it through the pumps which charge the accumulator. He proposes an accumulator so loaded that the ram ascends with the pressure due to the descent of the lightest load. Two similar forms of apparatus are provided, each consisting of a hydraulic cylinder, in which works a piston, with a weight on the top of its rod. The waste water from the hydraulic cranes and lifts (being under the pressure due to the weights of their descending rams) is admitted below the piston in one of these cylinders, and the piston rises until it reaches the top of the stroke, when it opens communication with the other cylinder, the piston in which also rises. Communication in the meantime has been opened between the top and bottom of the first cylinder, and at the same time between the cylinder and the high pressure main supplying the cranes and lifts. Owing to the difference of area for the water to act upon on the upper

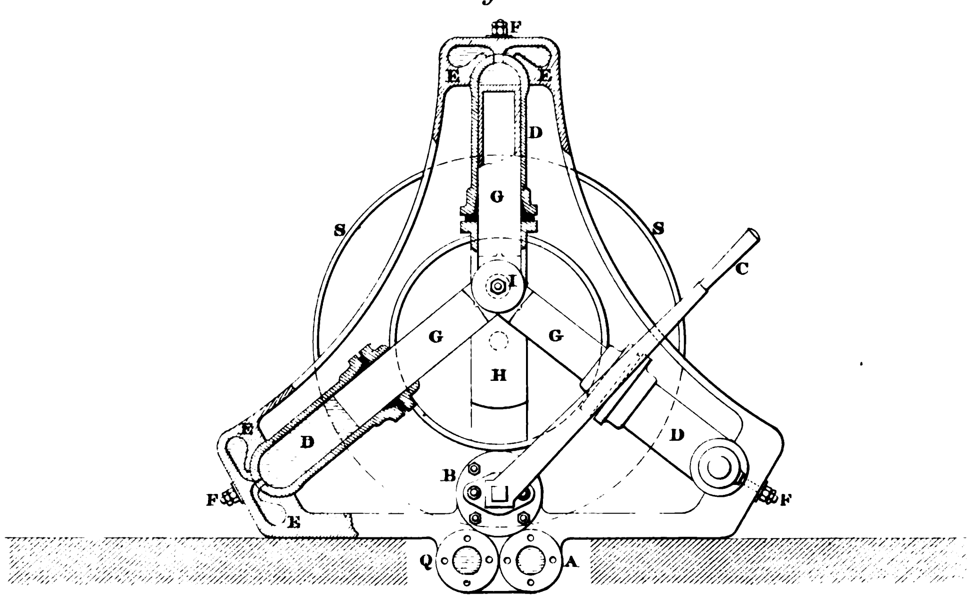
and lower sides of the piston, the pressure required to raise the piston is less than the pressure under which the water is forced out of the cylinder when the weight descends.

Mr. Robert Mills has devised a crane to meet the case of variable loads. The slewing cylinders which turn the crane (by chains passing round a drum fixed on the crane post) are utilised for the further purpose of aiding in lifting the weight. Four cylinders are generally employed, namely, the lifting cylinder, two turning cylinders (these three having rams), and a fourth, called the lowering cylinder. The latter has a piston, the rod of which is attached to the pulley-frame on the head of the ram of the lifting cylinder. The turning-chain passes round the turning-drum over the sheaves of the turning-rams as usual, but the ends are attached to the lifting sheave frame, so that the turning-rams can be used as auxiliaries to the lifting-ram, or independently of it, to lift light loads. In lifting a load by means of the turning cylinders, the water is admitted to both turning cylinders, and if the load is not too heavy to be lifted by these cylinders, the lifting sheave frames (with the ram and piston of the lifting and lowering cylinders) will be moved. The lifting cylinder and front end of the lowering cylinder will draw water by suction from the exhaust pipes. The water from the back of the piston of the lowering cylinder is discharged into the front end of the lowering cylinder (or into the lifting cylinder) along with the water from the exhaust. If the power of the turning cylinder is insufficient to raise the load, then the water is admitted to the front end of the lowering cylinder to assist in lifting, when, if the load is not too heavy, the lifting sheave frame with the ram and piston of the lifting and lowering cylinders will be moved. The back of the lowering cylinder discharges its water into the lifting cylinder, and the deficiency, if any, in the lifting cylinder is made up from the exhaust as before. Should the turning and lowering cylinders be of insufficient power to lift the load, the water is admitted to the lifting cylinder, but is shut off from the lowering cylinder. If the power that is now applied is sufficient to lift the load, the front



LONGITUDINAL SECTION

Fig: 2.



END ELEVATION.

Scale 1/16 th

Ins. 12 9 6 3 0 1 2 3 4 Feet

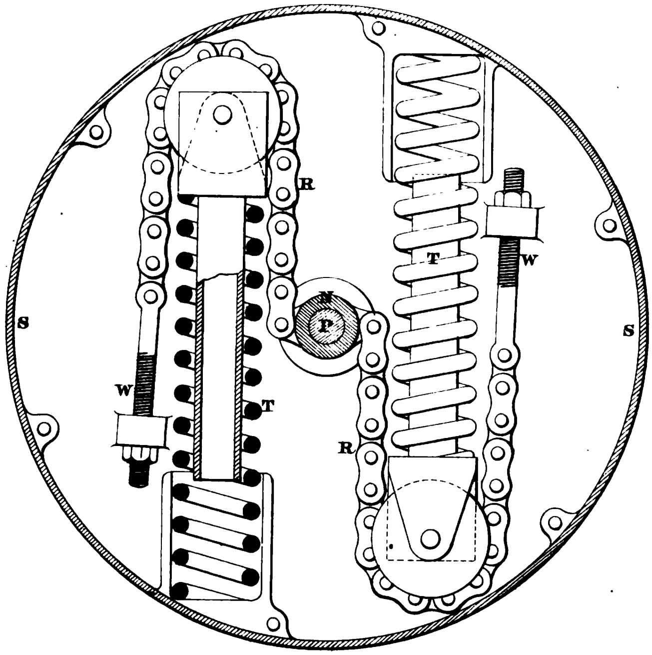
Thos Kell & Son, Lith.

end of the lowering cylinder will draw water from the exhaust. The back end of the lowering cylinder will then discharge its water (against the pressure) into the lifting cylinder. Should the power applied be insufficient to lift the load, the pressure will be readmitted to the front of the lowering cylinder, which will now act in conjunction with the turning and lifting cylinders. The back end of the lowering cylinder (as before) discharges its water, against the pressure, into the lifting cylinder. Should the power applied as last described be insufficient to lift the load, then the water from the back end of the lowering cylinder is allowed to discharge into the exhaust, when the full power of the crane will be exerted. When the crane is lowering the load, the water from the lifting cylinder is allowed to discharge itself into the exhaust, when the load (if heavy enough) will force the water from the front of the lowering cylinder into the pressure pipes. If the load is insufficient to lower in this manner, then the pressure is admitted to the back of the lowering cylinder, and the water from the front of that cylinder is discharged into the pressure pipe. The water from the lifting cylinder is also discharged into the exhaust pipes.

Mr. John Hastie adjusts the consumption of power in water-pressure rotary engines by varying the stroke of the piston. He accomplishes this by arranging the crank-pin so that it has a variable throw, and he also does the same by employing a link, as in the expansion gear of a steam-engine. An example of it as applied to a low pressure hoist is shown on Plates 10 and 11, which are taken from the *Proceedings of the Institution of Mechanical Engineers.* A is the inlet pipe, from which water is admitted through the cock B, under the control of the handle C. When the handle is in its extreme positions, the cock B acts as a reversing valve, and when the lever is vertical the cock acts as a break, owing to both parts of the cylinders communicating with the exhaust Q, the pipe of which contains water enough to fill the three cylinders D. Each cylinder alternately forces water into, or draws water from, the exhaust pipe Q. Two communication-passages are formed in the framing

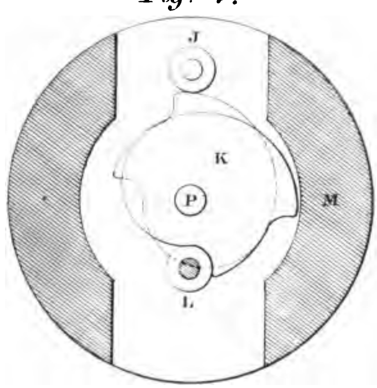
between the cock B and each of the cylinders D, the termination of these passages being at E (fig. 2). The oscillation of the ends of the cylinders serves instead of valves to admit the water. By the eye-bolts, F (fig. 2), the cylinders are held and adjusted, and working in them are trunnions that are cast on the cylinder ends. The rams G act direct on the crank pin I, formed on a sliding frame H, and by it the stroke can be adjusted. The double cam K (fig. 4) works between two plates, forming the frame H (figs. 1, 2, 5, and 6). To the outer plate are attached two small steel rollers, L and J, rolling on the outer and inner halves of the cam. Figs. 5 and 6 show the action of the frame H. This slides within a disc M, which is keyed on the hollow shaft N (fig. 1). The cam K is keyed on the barrel shaft P, fig. 1 (the working shaft of the hoist), passing through the hollow shaft, N. On the latter are formed two wings, to which are attached chains, R (fig. 3). To the barrel shaft P is keyed the spring case S, containing the two springs TT. When the engine is at rest, and without load on it (the crank-pin being then on its shortest throw), the springs TT have pressure put on them by the screws WW only sufficient to hold the roller J against the nearest end of the curve of the inner half of the cam. The pressure also prevents any change in the position of the crank pin, should the engine be running without any load. When a weight has to be lifted, the hollow shaft N at first turns and winds up the chains, compressing the springs, until their resistance balances the pressure put upon them by the weight, whereupon the spring-box S begins to revolve, carrying round with it the shaft P, thus raising the weight. The relative positions of the two shafts have now been altered from those shown in fig. 3, the rollers having moved along the curves of the cam, by which the position of the sliding frame has been shifted, and the crank-pin has been given an increased stroke proportional to the load that is being lifted. On the weight being removed, the springs open, causing the roller L to bring the crank-pin to its shortest throw. The variation of the stroke is thus automatic, and causes the consumption of water to vary with the work

Fig: 3.



SPRING CASE.

Fig: 4.



DOUBLE CAM AND ROLLERS

Fig: 5.

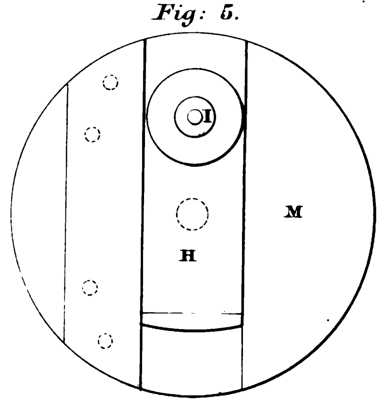
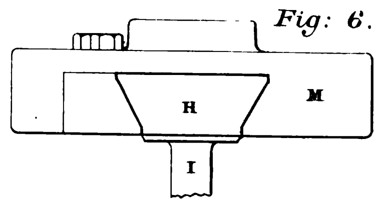


Fig: 6.



CRANK-PIN SLIDE

Scale 1/8th

Thos Kell & Son. Lith.

that is done, instead of the consumption being invariable, as is the case with the ordinary rotative engine.

When this arrangement is applied to the high pressures from accumulators, the springs are not employed, but water rams are substituted which are connected, through the centre of the barrel shaft P, with the supply pipe. The chains in this case are wound on cams, instead of on the shaft; greater power is thus required to force back the rams in proportion, as the chains act at an increasing distance from the centre of the shaft. Experiments have been made with one of these engines, which was applied to work a hoist with a 22-feet lift, and with a water-pressure of 80 lbs. to the square inch. The results are given in the following table:—

Weight lifted in pounds	Chain only.	427	633	745	857	969	1081	1193
Water used in gallons	7½	10	14	16	17	20	21	22

Whilst recognising the soundness of the principle of endeavouring to adjust the consumption of water to the work that is actually performed, some of the ingenious arrangements referred to involve the sacrifice of simplicity, and produce complications which it is most desirable to avoid. Even where means are provided for varying the power, the man in charge of the machine cannot be depended upon to employ them. In machinery for working wharf, coal, or barge hoists, where the weights to be lifted are great, and where the variation between the heaviest load and the lightest load is considerable, it is an advantage to put in one or more cylinders, or to make other arrangements for economising the water. In such machines no objectionable complications are involved, as is the case in smaller and lighter appliances.

For heavy cranes also variable power should be applied, unless in situations where the crane is very seldom used. Then the interest on the additional outlay of capital is not recouped

by any saving of water. In small cranes which are constantly worked, and where great rapidity is required, the double power is seldom of service. The men will often omit to work with the low power, as they find they can get so much more speed out of the higher power, when lifting light loads. The cost of the water thus wasted is comparatively so small that it does not pay to complicate the machinery in order to avoid the waste, especially as the addition of variable powers involves more attention on the part of the men, and more wear and tear to the machine. In working capstans it is found that, in hauling trucks, the men often prefer to put on the full pressure at the outset, to give a quick start to the load, and then to let the load run some distance, instead of using low power and keeping it on for a longer distance. It is found that the men are able to do the work better where they can start the waggons quickly, and with a good speed, than they can by using a lower power, and by keeping it on more continuously through the distance.

HYDRAULIC PRESSES AND LIFTS.

The Bramah Press is a practical application of the law of the equal transmissibility of fluid pressure, by which a force that is exerted by a small ram on one unit of water-surface is capable of being exerted over any number of units of water-surface in direct communication with the cylinder which contains the water that receives the initial pressure.

In presses of small diameter the calculation of the thickness, and the proportioning of the metal round the orifice admitting the water, have been matters of no difficulty, but the gradual increase in the size of the presses to meet the development of the use of hydraulic power has involved new arrangements of construction.

In determining the thickness of hydraulic cylinders, it must be borne in mind that, except where the thickness of metal is small compared to the radius, the conditions of strain on the

inner and outer radius of the metal are not the same, so that it must not be assumed that the thickness can be increased in direct proportion to the strain.

For thin cast-iron cylinders—

P = pressure in lbs. per square inch.
R = internal radius in inches.
T = thickness in metal.
C = coefficient for cohesive strength of the metal.

Then

$$\frac{T}{R} = \frac{P}{C}$$

When C = 16,500 lbs. for the bursting tension.
= 5,500 lbs. for the proof tension.
= 2,750 lbs. for the working tension.

The expression to find the thickness is generally taken as follows:—

$$T = \frac{P \times R}{C - P}$$

Where C = 16,500.

Where ordinary cast iron is employed, it was formerly considered that, beyond a thickness of 10 inches, a press was not increased in strength by adding to its thickness (the working pressure being taken at about 4 tons to the square inch). The introduction by Sir Joseph Whitworth of the principle of compressing molten steel, led to the construction of a press made from that material, which was 30 inches internal diameter, 5 inches thick, and has stood a working pressure of 3 tons to the inch. At Sheffield, steel presses are made 20 feet long, with 18-inch rams working at a pressure of 8 tons per inch, or over 25 tons per inch strain on the steel. Special care has to be taken to strengthen the inlet orifice. With ordinary cast iron, the practice has been to resort to hooping, when presses of large diameter, and subject to high pressure, are required. The external part of the cylinder ought to be placed in a condition of initial tension diminishing through a neutral point until a condition of compression exists. At the works of the Manchester Packing Company a press, 20 inches in diameter, hooped with

wrought iron, works at 2 tons per inch. Mr. P. R. Jackson mentioned at the meeting of the Institution of Mechanical Engineers in 1874 that some forty years before that date he had constructed an hydraulic press, with cylinders 21 inches in diameter, by employing a very thin shell of rather hard cast iron, and surrounding it with two or three sets of hoops or rings of wrought iron or steel, to give the requisite strength, and that by lining the cylinder with copper or brass it was rendered watertight under very heavy pressures.

In some cases where hydraulic presses are employed, the maximum pressure is not required to be exerted throughout the whole stroke. Many expedients have been resorted to by which a variable pressure is obtained. One means of effecting this is by connecting a low pressure accumulator during the first part of the stroke, and afterwards connecting with a high pressure accumulator.

Mr. Wilson of Patricroft has designed a combined high and low pressure pumping-engine to pump water at two pressures. The engines at starting run quickly, and continue to pump water at a low pressure until the resistance of the goods that are being pressed overcomes the low pressure engine, which then stops, and the high pressure engine continues and completes the pressing. He improved on this by employing three rams instead of one, and by working with one uniform high pressure (the low pressure engine being dispensed with). Water is admitted from the pumps to the centre cylinder only, which does the first work of pressing, and when the resistance balances the pressure of the centre ram the water is pumped into the outer cylinder as well, by which the pressure of the three rams is brought to bear. In case a series of presses are continuously at work, the water-pressure can be delivered direct from the pumps without the intervention of an accumulator, which, however, is of great service where there is intermittency of action.

Mr. Tweddell devised an intensifying apparatus for cotton-pressing, the principle of which consisted in placing two low pressure steam cylinders beneath two smaller hydraulic cylinders.

The ironwork of the Britannia and Conway tubular bridges was put together on the shore, and each tube was floated out on pontoons to a point which was vertically underneath its final position, when it was raised by means of hydraulic presses to the top of the pier and abutment upon which it was to rest. After several proposals had been discussed, it was finally decided that each tube of the Conway bridge should be raised by a single press at each end, the weight to be raised being 630 tons. The presses were placed at the top of the abutment upon cast iron girders, which spanned recesses that were left in the masonry for the ends of the tube to pass through. Upon the top of each press was a cast iron crosshead from which two suspension chains hung, the lower ends of which were made fast to the end of the tube. The internal diameter of the press was 20 inches, the thickness was $8\frac{3}{4}$ inches, and the stroke was 6 feet The diameter of the ram was 18 inches, and it was cast hollow. It was guided vertically by two 6-inch wrought iron guide-rods fitted in a socket at the top of the press, and keyed above into a cast iron girder built in the masonry. The piston of the force-pump was $1\frac{1}{16}$ inches in diameter, and its stroke was 16 inches. The pumps were worked by a steam-engine, having a 17-inch cylinder with 16 inches stroke. To lift the press 6 feet, the engine made 1018 strokes, or 34 per minute.

The weight of one of the tubes of the Britannia Bridge was considerably greater than that for the Conway Bridge, the total weight to be lifted being 2065 tons. As the original presses were not powerful enough for this load, another press was made. The two original presses were placed at one end of the tube to lift 921 tons, and the new one was placed at the other end of the tube to lift 1144 tons. This last press had a ram 20 inches in diameter, the cylinder having an internal diameter of $2\frac{1}{2}$ inches, and a thickness of metal of 6 inches. The weight of this press was 13 tons.

Considerable difficulty was experienced in casting these presses, and in rendering them watertight under the pressure. When a press was found to leak, the iron was well hammered,

and a thick gruel of oatmeal and sal ammoniac was forced into the press, the filmy particles of which were mechanically fixed in the pores by the corrosion produced by the sal ammoniac. The large press failed after lifting the tube 24 feet, the bottom separating from the body. After this accident another press was made, but it was not thought advisable to increase the thickness of the metal. Great pains, however, were taken in carefully selecting the iron to obtain the soundest possible casting.

In 1857 Mr. Bidder proposed to construct a graving-dock in connection with the Victoria Docks, which had just then been constructed. The experience that had been gained in raising the tubes of the Britannia and Conway tubular bridges suggested the construction of an hydraulic lift graving-dock. Mr. Edwin Clark was consulted by Mr. Bidder, and ultimately carried this out. The object to be attained was to enable vessels, that required overhauling or repairing, to be brought over a pontoon on which they could be raised and transported to a berth for repairs. The site adopted admits of a direct entrance from the docks, with a permanent water-level of 6 feet, this being the maximum draught of the pontoons. There are eight pontoon berths, each being 60 feet wide, and from 300 to 400 feet long, separated by jetties for workshops. The lift consists of two rows of cast iron columns, sixteen in each, with a clear space of 60 feet between the rows. The columns are 68 feet 6 inches in length, 5 feet in diameter at the base, and 4 feet in diameter above the ground, below which the columns are sunk 12 feet. They are 20 feet apart, and are placed on each side of the lift pit in 27 feet of water. They act as guides for the crossheads of the presses, but carry no weight. They are covered by caps, and a wrought iron platform runs the whole length of the dock on each side to connect them together. Each column encloses a hydraulic press 10 inches in diameter with a 25-feet stroke, having solid rams. A boiler-plate crosshead 7 feet 6 inches long is carried on the top of each ram, and from the overhanging ends of this crosshead are suspended,

by wrought iron bars, two iron girders 65 feet long, which extend across the dock to the corresponding column and press on the opposite side. A wrought iron platform is thus formed with 27 feet of water over it, when the rams are down. The rams have an area of 100 circular inches, the water-pressure being 2 tons per inch. This gives 200 tons lifting power for each press, or 6400 tons for the whole. Deducting 620 tons dead weight of working parts, the load that can be raised is 5780 tons. The presses are arranged in three groups, forming a tripod on which the pontoon is carried. The presses in each group are connected, and as each group is independent of the other two, the position can be either maintained level, or at an inclination. Any pair of presses can be cut off in the valve room, and, except in the case of the largest class of vessels, one or more of the end pairs is generally out of use. The water is supplied to the presses through a half-inch pipe from twelve force pumps worked from a 50 HP engine.

In 1872 the Government of Bombay had another Hydraulic Lift Graving-Dock constructed by Mr. Edwin Clark, which was an advance upon the original Victoria Graving-Dock Lift. This is constructed at Hog Island, Bombay, at a point where there is a depth of from 50 to 60 feet of water, and a maximum rise of tide of 16·7 feet, and an ordinary rise of 14 feet. The length of the dock is 350 feet. There are two rows of cast iron columns, having a clear distance of 88 feet between them. There are eighteen columns in each row, their distances apart gradually increasing from 18 feet in the middle to 24 feet at the ends. Each column is 7 feet 6 inches in diameter at the base, and 6 feet 6 inches above, the length being 100 feet. They are placed in pits sunk in the soft rock, to which they are bolted and concreted, 36 feet of the column only appearing above high water. In each column two hydraulic presses are enclosed with solid rams, 14 inches in diameter and having a stroke of 35 feet. Each press is in one piece, 35 ft. 6 in. long and it weighs 14 tons. Two long girders, 2 ft. 7 in. deep, run along the top of each row of columns, and on these girders

are four 25-ton travelling cranes for raising the presses, if repairs are required. At the end of each row of columns is fixed a fender column, 10 feet in diameter, filled with concrete. The pontoon is 380 feet long, 85 feet broad, and 9 feet 6 inches deep at the outside, and 6 feet 6 inches in the centre. It weighs 1610 tons, and is capable of raising 6500 tons. It is divided into 36 water-tight compartments, in the bottom of which is a valve. The presses are divided into three groups, the water being admitted at the top of each press. The presses were tested to 1·7 tons per square inch. The rams have an area of 196 circular inches, so that with a working pressure of a ton to the inch a weight of 14,000 tons can be lifted, which is done in half an hour. This is in excess of the load ever required to be raised, which is considered to be 9000 tons, of which 6000 tons is the weight of the vessel.

A similar hydraulic lift dock was constructed at Malta, and it was opened in January 1873, by docking H.M.S. *Cruiser*. The length of this dock is 340 feet, and the clear width is 62 feet. It is capable of lifting vessels up to 3000 tons register, and drawing 21 feet of water.

Mr. Leader Williams adopted hydraulic power in lifting barges so as to connect the river Weaver with the Trent and Mersey Canal at Anderton. The difference of level being 50 feet, the process of locking was tedious and expensive. The plan that was adopted consisted in constructing a wrought iron aqueduct by which the canal was brought to a point where the barges could be best raised and lowered to and from the river. Mr. Duer (who was resident engineer of this work) described it fully in a paper read before the Institution of Civil Engineers in 1876.

Fig. 12 shows a side elevation of the general arrangement of the works. Fig. 13 is a cross section of the lift and aqueduct. The wrought iron aqueduct is 162 feet 6 inches long, by 34 feet 4 inches wide, in three spans of 30 feet, 75 feet, and 57 feet 6 inches. It is divided into two channels by a central web, the depth of it and of the sides being 8 feet 6 inches. The water is 5 feet 3 inches deep, and with the aqueduct gives

a total weight of 1050 tons. This weight is partly supported by the columns A, which rest on cast iron cylinders con-

ANDERTON LIFT.
SIDE ELEVATION.

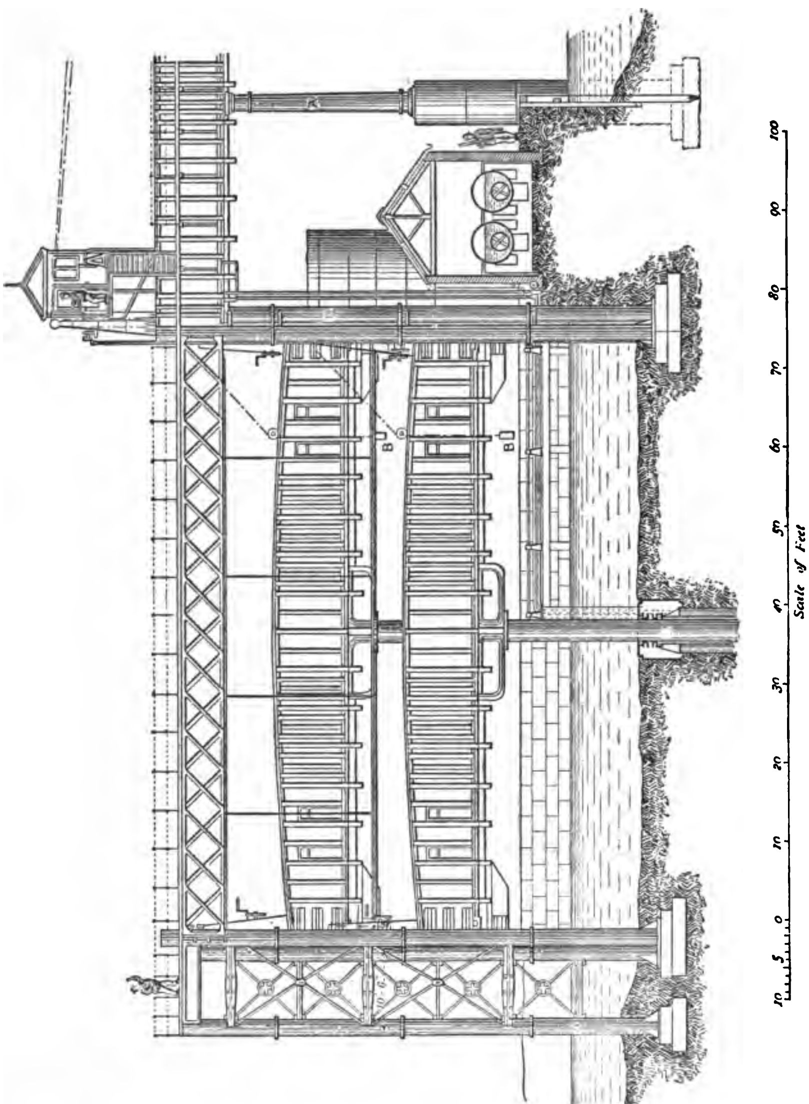


FIG. 12.

taining concrete. These are carried on masonry foundations built on piles. A water-tight connection is obtained by bolt-

ing the wrought iron bottom-skin of the aqueduct upon a cast iron bed-plate that is built into the masonry with a layer of red lead between. The outer edges are caulked with wooden wedges, and the sides are run with Portland cement. Each end of the

ANDERTON LIFT.

CROSS SECTION.

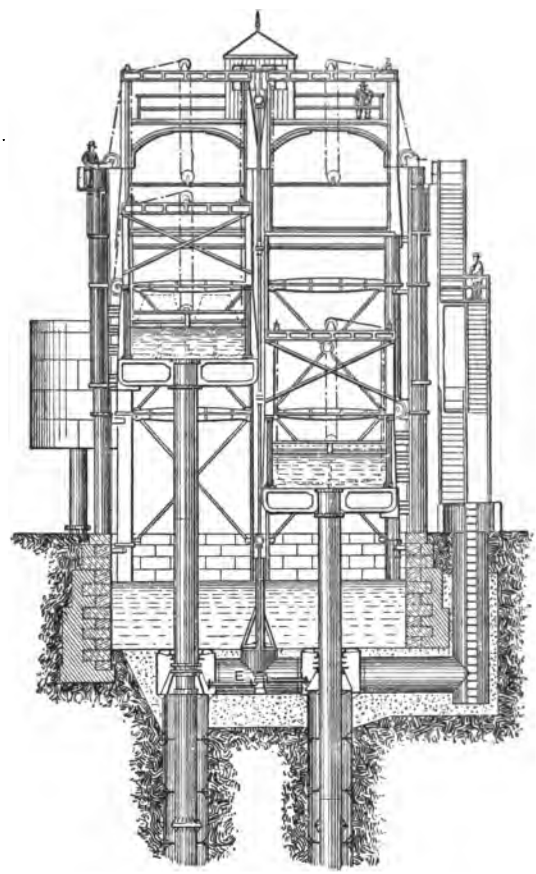


FIG. 13.

aqueduct is fitted with wrought iron lifting-gates, made water-tight by indiarubber strips fitted between them and the aqueduct. Each gate weighs 27 cwt., and is counterbalanced by weights, B (fig. 12). The lifting of a gate can be effected in a minute and a

half by a crab. The gate is raised 7 feet 6 inches clear of the water, which enables the highest barge to pass under. The lift is double, so that by means of two troughs, with their floating barge load, the upper one, in descending, can be adjusted by the admission of water, so as to raise the lower one. These troughs are each 75 feet long by 15 feet 6 inches wide. The lighter barges hold 30 tons, and the heavier 100 tons of goods. The troughs have lifting-gates at their ends like those on the aqueduct. One central vertical ram, 3 feet in diameter, supports each trough, whose weight (with the water and barge) is 240 tons, which is equivalent to a pressure of $4\frac{3}{4}$ cwt. per square inch of the ram. The rams are raised by presses controlled by an equilibrium valve for opening and closing communication between them. A 5-inch pipe connects these presses, and a 4-inch pipe conveys the water from the accumulator to the presses. One man in a valve house at the top of the aqueduct works the lift by means of shafting and gearing. When a trough descends into the pit, it is immersed fully 5 feet. The depth of water while the trough is being lifted, however, must not be more than 4 feet 6 inches, the extra water being drawn off by syphons which dip into the water while a trough is descending. The air within the syphon is driven out into the trough by its shorter leg, which nearly fills the trough with water. When it is again lifted, the syphon draws water (owing to the partial vacuum within it) out of the inside of the trough, and thus acts automatically. Each trough can, if necessary, be lifted separately by the engine and accumulator, but this occupies half an hour, whilst the double lift is made in from two to three minutes with a 10 HP engine. A single lift could only take two barges up, or bring two down in eight minutes, with an engine of six times the power required for a double lift.

The abstraction of 15 tons of water from the canal (representing a layer of 6 inches over the bottom of the trough) provides the chief means required for raising a barge. The remainder of the power (about one-twelfth) is obtained from a small steam engine and accumulator. The double-lift arrangement enables

expedition and economy to be secured, as each press alternately utilises the weight of the trough, which rests upon it, to raise the other trough from the low to the high level.

A saving of water also is effected as compared with locking, inasmuch as only 15 tons are used at each operation of raising a barge, whereas with a fall of 51 feet through a chain of six locks, a much larger quantity would be wasted. Under the most unfavourable circumstances (such as, *e.g.*, two similar barges having to pass each other through locks with this fall) the column of water taken from the upper level would be equivalent to the area of one lock multiplied by the total fall. If, however, a series of barges were arranged to follow each other in the same direction, less waste would ensue. If six barges were to ascend with all the locks empty, the first would take five lockfuls, and the other five would take one lockful each from the upper level, making ten lockfuls for the ascending barges. A similar number of descending barges would take eleven lockfuls of water, making twenty-one altogether, or 175 feet, whereas the lifts would require six layers of water each 6 inches deep, or 3 feet, which is only 1·7 per cent. of that which would be used for locking. This lift is capable of taking eight barges up and eight down in an hour. Assuming eight to be laden with the average load of 25 tons each, the lift can thus transfer 12,000 tons per week, at a cost of 2·16 pence per ton. The parliamentary tolls were as follows:—Per ton for all goods, 1d. For each laden barge, 1s. For each empty barge, 2s. 6d.

In 1882, one of the hydraulic cylinders of the Anderton lift was fractured during its use. At the time of the accident one of the troughs had been raised to the top, and the aqueduct gate was partly open, when the trough fell, owing to a side of the press having blown out. This occurred before the six inches of water had been admitted to it, so that it was empty, and the pressure on the cylinder of the press was calculated to be 532 lbs. per square inch. On applying a pressure of 800 lbs. to the second press, it cracked through the hole for the inlet pipe. The fractures in both presses were similar in position and

character. The part that failed in the first press was not the press proper, but the upper casting, or press head, with the opening for the 5-inch supply-pipe. The continuity of the circumference was practically destroyed at that part, a greater strain being imposed thereby on the surrounding metal, and, in addition, the thickening of the cylinder at the inlet caused unequal contraction on cooling, and consequently rendered the casting less sound. From an investigation that was made by Mr. Edwin Clark it appears that the following were the conditions at the time of the accident:—The water-pressure, as before stated, was 532 lbs. per square inch. The diameter of the ram was 37·5 inches, and the thickness was 2·5 inches Then by the expression—

$$S = \frac{P.D}{2T}$$

(where S is the tensile strain per square inch of section of metal, P is the pressure per square inch, D is the diameter, T is the thickness), it will be seen that S is 1·78 tons per square inch. In the case of the second press that was tested to bursting, P = 800 lbs. per square inch. Hence S = 2·68 tons per square inch, which is below the strain to which cast iron is usually subjected. If the press had been a simple cylinder, it should have borne—

$$\frac{2 \times 2{\cdot}5 \times 7 \times 2240}{36} = 2090 \text{ lbs. per square inch,}$$

as compared with 800 lbs., which was the actual pressure when it burst.

The failure of the Anderton press caused inquiry to be made into the circumstances of the case, as similar lifts were being proposed for other places, notably in France, on the Neuffossé Canal at Les Fontinettes, near St. Omer, and in Belgium, at La Louvière on the Canal du Centre, near Mons. The observations that have been made to determine the construction of the presses for the Louvière Canal lifts, are interesting and important. It was originally intended that the press should be of cast iron, 6 feet 8 inches internal diameter, with metal 4·72

inches thick. The pressure in the cylinder being 28 atmospheres (about 420 lbs. per square inch), the extreme tension would have been 1·65 tons per square inch. This was considered a safe load for the Belgian cast iron, which bears a tensile strain of 11·43 tons per square inch. The Terre Noire Steel Company of St. Etienne, France, suggested a press of cast steel, constructed in the same manner as an ordinary cast iron press, but of less metal. Some of these rings were cast and tested. One of them was kept under a pressure of 46 atmospheres for two hours, and proved perfectly watertight. Trial bars cast at the same time broke at 31·16 tons per square inch, with an elongation of 8·6 per cent. Another ring, chosen haphazard, was tested. At 50 atmospheres an elongation of ·157 of an inch was measured. On removing the pressure, the press returned to its original dimensions. At 75 atmospheres the elongation was ·197 of an inch, and at 80 atmospheres the press suddenly failed; on examining the fracture a fault 5 inches long, and extending nearly through the whole thickness of the metal, was seen. It was due to a scale from the mould becoming detached, owing to the high temperature of the casting, though, judging from the trial bars, the press should have withstood 240 atmospheres. Owing to this failure it was determined to abandon this form of construction.

Messrs. Cail of Paris next proposed a press of steel plates bent into a cylindrical form (like a boiler), with rivetted butt joints having internal and external cover-plates. A trial length was built up in rings 6 feet 3 inches high, with covering-rings at the joints. The steel plate was 1·02 inches thick, with a working tension of 7·17 tons per square inch. Although the rolled plate would stand 38 tons, the weakening due to rivetting reduced the margin of safety, and the joints could not be made watertight. A trial length leaked badly under 30 atmospheres, and at 35 the pumps could not make up the leakage. Ultimately, the press cracked through the cover-plate, and some of the rivets started, at a pressure which could not be definitely ascertained, but was between 48 and 70 atmospheres.

While these trials were going on, Messrs. Clark & Standfield had been directing their attention to the placing of steel hoops round the cast iron presses. The practical difficulty of getting the hoops over the flange of the press presented itself, and it was decided to make the hoops at the joints with flanges like the tire of a wheel. To prevent this flanged tire from being dragged off, a small projection was left on the body of the press, the heated tire was then passed over this, and, in cooling, it fitted tightly behind it. Messrs. Clark & Standfield acted upon a long and interesting memoir by M. Kraft, chief engineer of the Société Cockerill, on the calculations for, and method of constructing, these presses, and a trial segment was made by

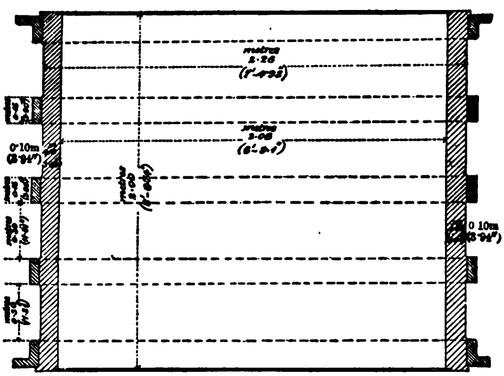


FIG. 14.

the Société Cockerill, as shown in fig. 14. This was tried under hydraulic pressure, the expansion being measured by a thin strip of metal put round the cast iron cylinder, and another strip round one of the steel hoops. The two ends of each strip were connected by means of a spring adjusted by a screw, and were also joined to the short ends of a kind of proportional compasses, set to a ratio of 12 to 1. By this means any slight opening of the ends of the strips, caused by the expansion of the cylinders, was shown twelve times its actual size on the long arms of the compasses. Owing to friction (which was, however, reduced to a minimum by lubrication) and other causes, the measurements were not absolutely correct, but the

instrument was found to be very sensitive and constant. A satisfactory trial took place in the presence of the ministers, and many Belgian and French engineers interested in the undertaking.

In addition to this trial, M. Génard (on behalf of the Ponts et Chaussées) and Mr. Lyonel Clark (on behalf of Messrs. Clark, Standfield & Clark) carried out a series of exhaustive trials on the segment, for the purpose of finding out as nearly as possible the conditions of the several portions of this composite construction under various strains. It is evident that the cast iron body is subject to a strain at the part covered by the steel coil entirely different from that to which it is subject elsewhere. Very many experiments were made, the pressure being increased gradually, and a measurement being taken at each increment of ten atmospheres. The mean of these, corrected for atmospheric temperature and other causes, was taken, and a normal curve plotted, which gave as the elongation on the cast iron between two coils, and elongations on one of the steel coils, the results shown by Table I. on p. 73.

Were the press a plain cylinder, it would be easy to deduce the tensions from these elongations, supposing the different coefficients of elasticity of the metal under the different tensions to be known; but in either case, before pressure was put on the press, the steel coil was already compressing the cast iron body to some extent. The tensions had, therefore, to be deduced in two ways, by calculation and by graphic means. The sizes to which the coil was bored, and the press turned, were accurately known, and a pressure which would compress the cast iron and elongate the steel coil until they became of equal length, was deduced. Although following different methods, both M. Génard and Mr. Clark obtained nearly the same results. M. Génard found the pressure existing between the coil and the press to be 14 atmospheres, whilst Mr. Clark found 13$\frac{3}{4}$ atmospheres. When considering the measured elongations, the tension on the cast iron body of the press is evidently relieved by this exterior pressure of 14 atmospheres, whereas the tension on the steel

TABLE I.

Actual Elongations of the Circumference.

	Pressure in Atmospheres.														
	10.	20.	30	35.	40.	50.	60.	70.	80.	90.	100.	110.	120.	125.	
Elongation between coils	·208	·573	·960	1·05	1·361	1·805	2·235	2·662	3·07	3·447	3·717	3·966	4·254	4·604	Millimetres.
	·0082	·0226	·0378	·0413	·0535	·0714	·088	·1047	·1209	·1358	·1464	·1655	·1675	·1812	Inches.
Elongation on coils . .	·134	·349	·578	·666	·884	1·24	1·618	1·974	2·318	2·696	2·992	3·242	3·475	3·766	Millimetres.
	·0053	·0137	·0228	·0262	·0348	·0488	·0637	·0777	·0913	·1061	·1177	·1315	·1368	·1482	Inches.

coil is increased to the same amount. They therefore found that the tensional strains were as shown in Table II. The strains at A are those on the cast iron press directly under the steel coil, and those at B on the steel coil itself.

It will be noticed that, with the interior pressure of ten atmospheres, the cast iron is still in compression, owing to the shrinking of the steel coils.

For that portion of the cast iron part of the press which does not lie directly under the steel coils, it was more difficult to calculate the tensions, for it was nearly impossible to find out to what extent the shrinkage of the steel coil influenced this part. It evidently lay between the maximum (that is, assuming this part to be as much affected by the shrinkage of the coil as the part directly under a coil) and the minimum, assuming the coil to have no influence. Table III. shows the results.

The ordinary working pressure of these presses is 35 atmospheres (517 lbs. per square inch). In this condition, then, the strain on the cast iron under a coil is 1·35 kilogrammes per square millimetre (·857 tons per square inch), for the cast iron between two coils, 3·175 to 3·475 kilogrammes per square millimetre (2·01 to 2·2 tons per square inch), and for the steel coil itself, a tension of 6·4 kilogrammes per square millimetre (4·06 tons per square inch).

The limit of safety fixed by the Belgian Government for cast iron under tension is 2½ kilogrammes per square millimetre (1·59 tons per square inch), and for the steel 7 kilogrammes per square millimetre (4·76 tons per square inch). It is evident, however, that although that portion of the cast iron which falls under the coils, and also for some distance on each side of it, is working under safe conditions, there is a portion which exceeds these limits. It was therefore decided by the Government that whilst accepting this form of press, they considered it desirable that a greater number of coils should be shrunk on, and it was eventually decided to make these coils continuous from top to bottom. The disposition is shown on Plate 12.

TABLE II.

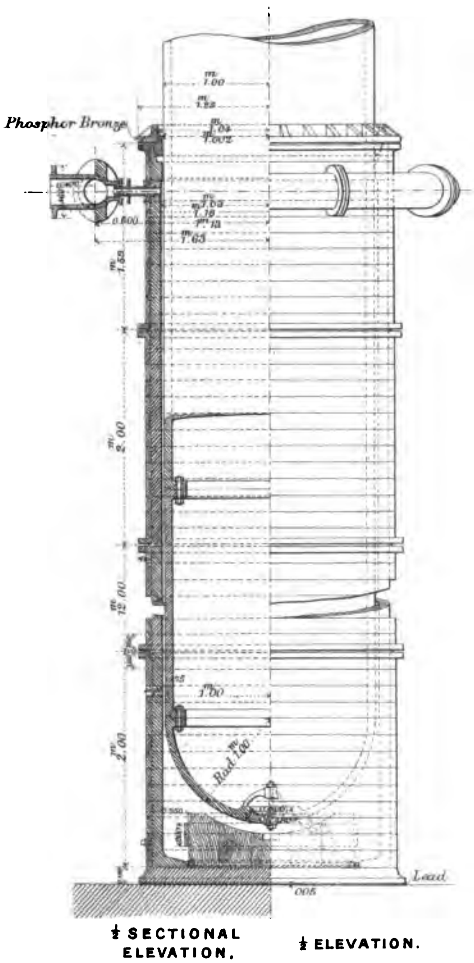
	Pressure in Atmospheres.												
	10.	20.	30.	35.	40.	50.	60.	70.	80.	90.	100.	110.	
A . .	−·6	+·18	·95	1·35	1·72	2·5	3·275	3·95	4·7	5·4	6·05	6·7	Kilogrammes per square millimetre.
	−·331	·1143	·6032	·8572	1·092	1·587	2·076	2·508	2·984	3·429	3·841	4·25	Tons per square inch.
B . .	4·33	5·15	6·05	6·4	6·85	7·7	8·6	9·45	10·15	11·29	12·15	13·03	Kilogrammes per square millimetre.
	2·73	3·24	3·84	4·06	4·32	4·89	5·46	6·00	6·57	7·17	7·71	8·25	Tons per square inch.

TABLE III.

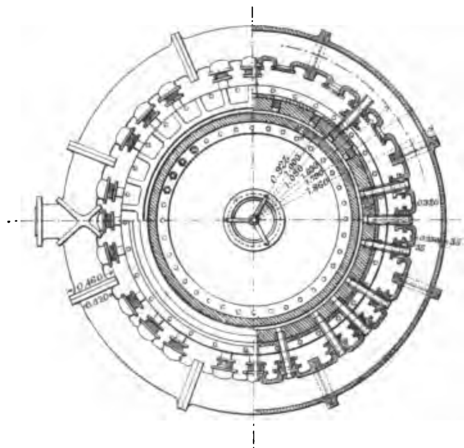
	Pressure in Atmospheres.												
	10.	20.	30.	35.	40.	50.	60.	70.	80.	90.	100.	110.	
Between	1·05	2·025	3·00	3·475	3·95	4·85	5·75	6·65	7·60	8·50	9·15	10·5	Kilogrammes per square millimetre.
	·667	1·28	1·905	2·2	2·51	3·08	3·65	4·22	4·83	5·397	5·94	6·67	Tons per square inch.
Coils	·57	1·7	2·7	3·175	3·65	4·55	5·43	6·33	7·23	8·1	9·0	9·9	Kilogrammes per square millimetre.
	·362	1·079	1·714	2·01	2·32	2·9	3·45	4·02	4·59	5·13	5·71	6·28	Tons per square inch.

In this press the shrinkage given to the steel coils is such that with the ordinary working pressure of 35 atmospheres (517 lbs.), no strain whatever falls on the cast iron. This only acts as a watertight lining, and gives stability as a column, the whole tension being received by the weldless rolled steel hoops. The press is, for convenience of manufacture and erection, constructed in segments, which are afterwards bolted together. The particular coil that terminates each section is constructed with a flange for this purpose. These coils are rolled out of soft Bessemer steel in an ordinary tire-rolling machine. They are 6 inches deep with 5-inch flanges, and in diameter do not exceed that of some express engine wheels. The Société Cockerill succeeded in rolling these in their existing tire mills without any extra preparations. The boring and turning have to be done very accurately, but all the pieces are of one size. The packing of these presses will be hemp and tallow, the same as at Anderton. The gland will be in phosphor bronze. The press itself reposes directly on a carefully levelled stone bed, which rests directly on the concrete at the bottom of the shafts. A sheet of lead ensures watertightness. The pressure is therefore taken directly by the solid earth, and there is no danger of that common accident, the blowing out of the bottom of the press. The feed inlet (on account of the form of the coils) has had to be considerably modified. It was found necessary to have at least 100 circular inches of feed-pipe. This it was impossible to obtain by a single pipe without cutting away some of the coils. Mr. Edwin Clark then devised the ingenious arrangement shown on fig. 1, Plate 12. A circular supply tube surrounds the press, from which project several smaller pipes, like spokes of a wheel, each one being a feed-pipe. The size of these pipes is 3 inches. The 6-inch coils, therefore, are not much weakened, and these particular ones being made thicker, the press is in reality not weakened at all. The supply has, moreover, the advantage of being remarkably even and regular, owing to the water entering at different points all round the press. The press is stayed against the sides of the well by

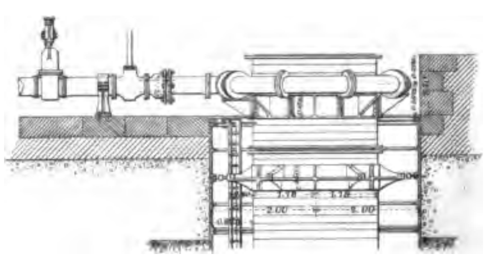
LA LOUVIÈRE PRESS. PLATE. 12.



½ SECTIONAL ELEVATION. ½ ELEVATION.



PLAN.



Thoˢ Kell & Son, Lith.

projecting brackets, furnished with adjusting screws. This is not a very necessary precaution, for the trough itself is so well guided that any deviation from the vertical line is impossible.

In designing these lifts, the principle of the Anderton lift was followed, varied, however, in one important point. It has been stated, when describing the Anderton lift, that the upper trough with its barge is made heavier than the lower one, by the addition of a layer of six inches of water, which forces the lighter one up. When the heavier one, however, enters the water at the low level, the displacement of the water diminishes its weight, and requires the action of a differential accumulator to complete the work, by supplying the power necessary to overcome the difference of weight, and to force the rising trough to its proper height. In these lifts this accumulator, with the engines, boilers, pumps, and labour, are dispensed with, by arranging the works so that the descending trough is received in a dry basin from which the low level water is excluded by a gate similar to that applied at the high level. This alteration in the design enables the descending trough to complete the operation of raising the other trough through the full stroke of the ram.

Plate 13 gives a general view of the hydraulic canal-lift at La Louvière. A similar canal-lift is being erected for the French Government at Les Fontinettes.

Sir John Fowler proposed in 1869 a scheme (which was further developed in 1884) for carrying on the traffic between England and France by means of steamers of very large size, upon which complete trains of carriages would be carried across the channel. By this plan a train arriving at Dover from London would be run on to the main deck of the steamer (the engine being of course first removed), upon which it would be carried over to Calais or Boulogne (at both of which places it was proposed to construct harbours), and then run off the steamer on to the railway, and so continue its journey to Paris. One of the many difficulties which had to be overcome in settling the details of the scheme was that of enabling the

LA LOUVIÈRE LIFT.

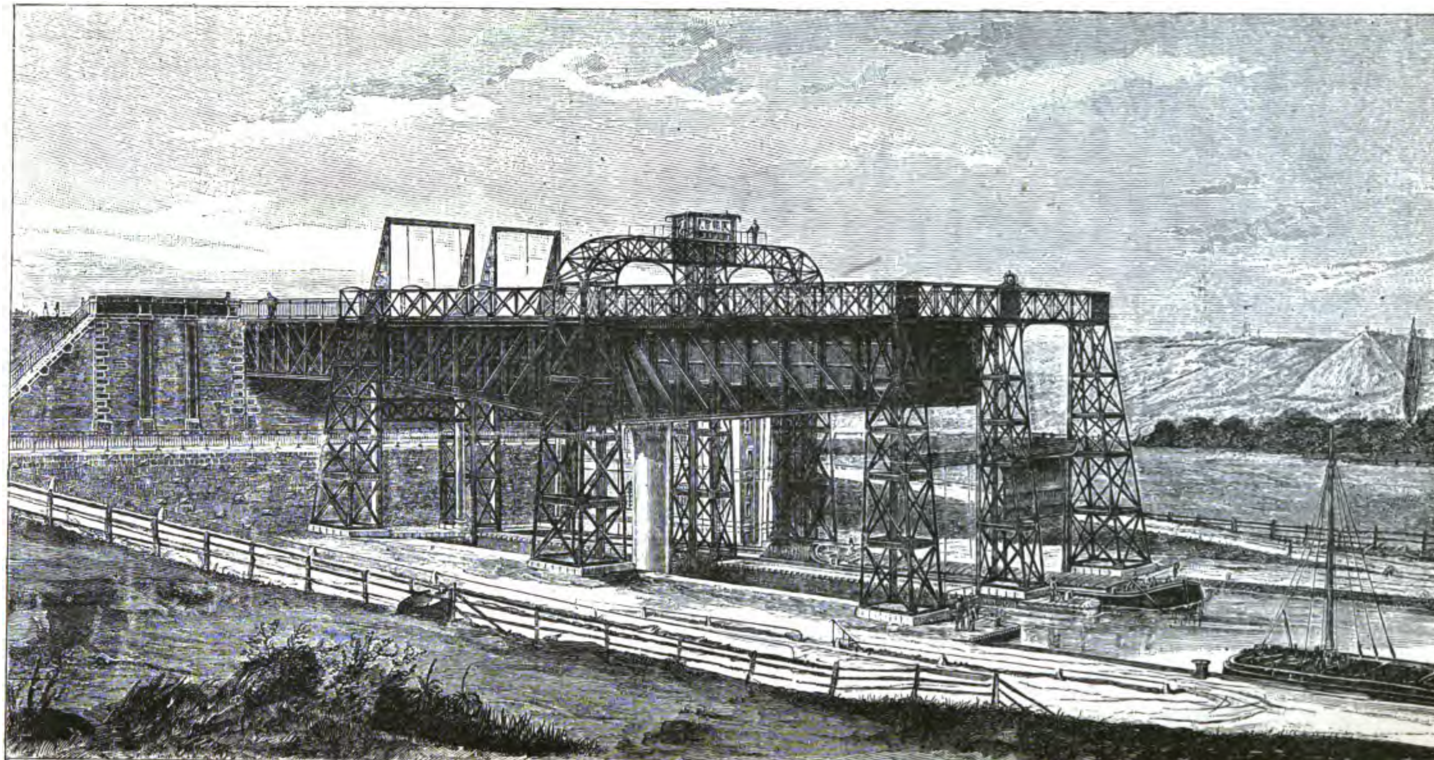


PLATE 13.

trains to be run on and off the steamers at all times of the tide, and the plan decided on was to construct hydraulic lifts which could be raised or lowered to the required level. The train on reaching the harbour would run directly on to one of these lifts, then be lowered to the level of the steamer waiting for it in a specially constructed dock, and finally move forward over a gangway on to one of three lines of railway laid upon the main deck. Each of these lifts was to be 300 feet long, and to be worked by three sets of hydraulic presses fixed above it, one set near each end, and one set in the middle. Each set was to consist of three presses placed transversely to the lift, sufficiently high to enable the train to pass underneath. These presses were to be supported by iron frames. The weight of each lift with its train would be about 200 tons, and the time required for raising or lowering a train would be one minute. It was intended to work the presses from an accumulator in a building placed at a convenient distance from them.

A cage raised and lowered on the top of a ram (the cylinder being sunk in the ground) is the simplest form of hoist. Provision in this case has to be made for a varying weight due to the altered condition of the load. As the ram rises, the head and pressure diminish, whilst the weight of the ram increases, as it is less and less immersed in the water. A counterbalancing weight is, therefore, required to lower the cage when empty, and to adjust the varying weights of the chain as the cage rises and falls, and also to balance the weight of the ram. The counterweight is usually attached to the chains connected with the cage, and passing over fixed sheaves at the top of the lift-framing. The amount of weight to be provided must be sufficient to balance the cage and the whole weight of the ram when at the top of the stroke, *minus* the weight of the chain which then assists the counterweight. When the ram is at the bottom of the stroke, the counterweight must balance cage, ram, and chain, the weight of the ram being then less than when it was at the top of the stroke, owing to water surrounding it. It will be seen that where the weight of a direct-acting ram is counterbalanced,

the ram is subjected to both tensile and compressive strains, according to whether the ram is being pulled by the counterweight or pushed by the water-pressure. If, on the other hand, the counterweights are omitted, the amount of water consumed to raise the load is greater in proportion to the useful work done.

The railway under the river Mersey (which was constructed by Sir James Brunlees and Sir Douglas Fox, and opened by the Prince of Wales in January, 1886), has in connection with it at each extremity hydraulic lifts for conveying passengers and their luggage from the deep underground stations at James Street and Hamilton Street to the daylight stations on the street level above. Particulars of these lifts were given in a paper read at the Institution of Civil Engineers by Mr. Rich (of the firm of Easton & Anderson, their constructors). Plate 14 shows the arrangement. The lifts at the James Street station have a stroke of 76·6 feet, and at the Hamilton Street station 87·7 feet. At each station there are three lifts independent of one another, each being capable of raising one hundred passengers at a time. The maximum load due to passengers is taken at 15,000. lbs. The lifts are direct acting, with rams of hollow steel 18 inches in diameter, with balance chains and counterweights. The ascending cage is 19 feet 6 inches long by 16 feet 6 inches wide and 8 feet 10 inches high. They are worked by low pressure water derived from a tank, aided by water pumped by steam-power direct into the lift supply-pipe. The tank to give the water-pressure is placed at the top of a block of buildings at each end of the tunnel. It is 15 feet 6 inches in diameter, 9 feet deep, and contains 10,000 gallons of water, representing storage for twelve journeys. The water is discharged by the descending cages into an underground tank, from which it is pumped back to the high level tank, the effective head in which is 176·5 feet at James Street, and 164 feet at Hamilton Street.

The several lifts are contained in rectangular vertical shafts, 21 feet long and 19 feet wide, partly excavated out of the solid

red sandstone, and partly in walls of brickwork. In the centre of each lift space, a boring has been carried vertically beneath the floor to a depth of 75 feet to receive the lift cylinders, which are of cast iron, 21 inches internal diameter, and 1½ inches thick, bolted together in 12-feet lengths. The rams are 18 inches outside diameter, and ½ inch thick, constructed of mild steel tubes in lengths of 11 feet 6 inches, and connected together by internal screwed ferrules 6 inches long and 15¾ inches internal diameter. The cage is guided and kept in position by four cast iron guide brackets (of a V shape) 16 inches long. From the side girders two chain pulleys, 4 feet 8 inches in diameter, are suspended. Between each pair of them is a counterweight weighing 7620 lbs., capable of being increased by smaller weights of 90 lbs. each to balance the lift. A large self-acting flap-valve admits water automatically to the lift cylinder from the exhaust, if the starting valve is closed too suddenly during the ascent of the lift. The stroke of the hand-rope, from full pressure to full exhaust, is 9 feet, which enables the starting and stopping to be effected quietly. Three 7-inch mains descend to each lift from the bottom of the supply tank, with the necessary valves to control the service. The speed is about 2 feet per second, and the average journey is accomplished in from thirty to forty seconds. The three lifts at each station are capable of working simultaneously, raising three hundred passengers in about a minute. The total cost of the six lifts with all machinery was about £20,000.

Hydraulic power is finding a large field for useful employment in the direction of working house lifts. Until recent years the form of lift to which hydraulic power was chiefly applied was that for raising and lowering loads, either by a platform placed on the top of a direct acting ram, or by a cradle attached to a chain or a wire rope, either passing over multiplying sheaves on a ram in an hydraulic cylinder, or wound over a drum worked by a rotary engine. Hydraulic lifts or hoists were formerly chiefly used in railway goods-yards and termini, docks, &c. In more recent years, however, the field of application has been extended

to offices, hotels, and private houses, where the height of the upper floors renders some mechanical appliance necessary.

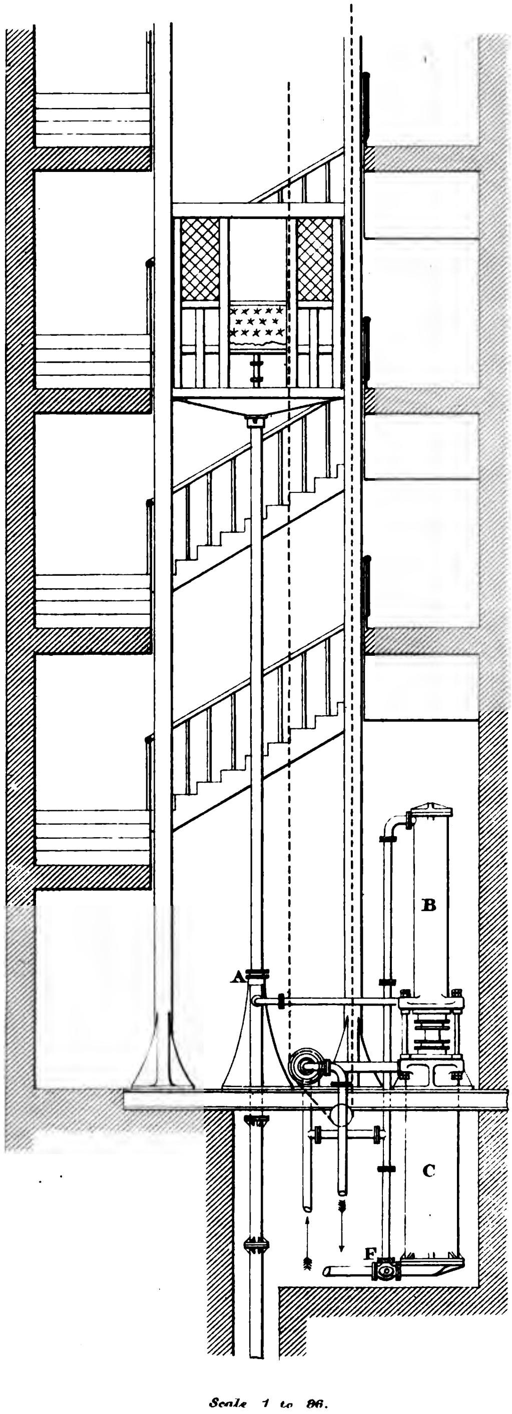
Messrs. Tommasi and Heurtvisé have devised a plan to balance the dead weights by means of a second hydraulic cylinder placed close to the lifting cylinder, and connected with it. The ram in this second cylinder is loaded so as to balance the lifting ram and cage when at the bottom. It has a larger area but shorter stroke than the lift-ram, and is continued of the same diameter, through a stuffing-box, to another cylinder above it. The pressure in this latter cylinder, acting on the ram, balances the lifting ram in the lifting cylinder with its cage when at the bottom. Counterweights serve to further balance the lifting ram as it rises, so that the pressure required to be applied to the lifting ram is only that which is necessary to raise the people in the cage, and the lifting ram is (as it should be) always in compression.

Mr. Ellington has designed an hydraulic balance lift for low pressures, which was explained to the Institution of Mechanical Engineers, and is shown by Plate 15.

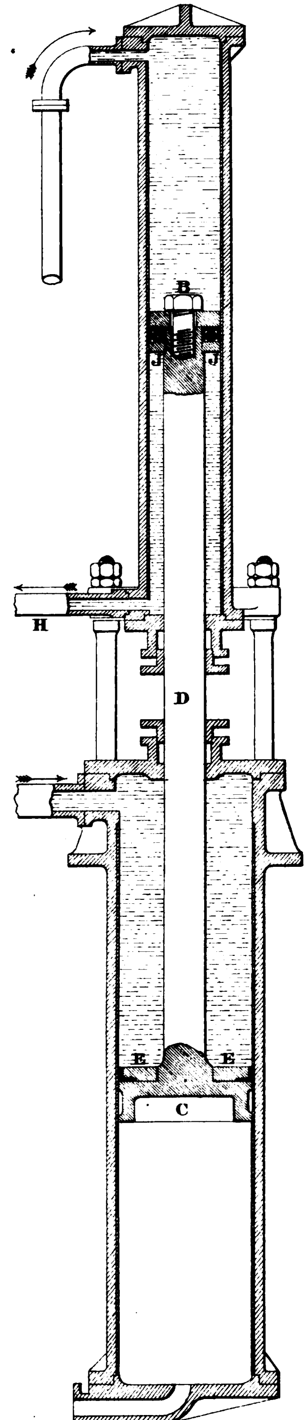
As in the previously described lift, the ram is always in compression, and the dead weight of ram and cage at bottom of stroke is balanced by means of a second cylinder B (11 inches diameter and 50 feet 6 inches stroke), which is in hydraulic connection with the lifting cylinder A ($3\frac{1}{2}$ inches diameter). The piston B has the pressure always on the upper side. The piston rod D of the second cylinder B is continued into a third and larger cylinder C ($21\frac{3}{4}$ inches diameter and 8 feet 2 inches stroke), and has a piston of a diameter calculated to give an annular space EE sufficient to lift the weight of the people in the cage, and to overcome friction.

When the cage is to be raised, water is admitted to the top of the third cylinder C, and the pressure exerted by it on the annular space EE, together with the constant pressure on the piston of the upper cylinder B, form a continued pressure which is exerted on the annular space JJ of the upper piston B, and is transmitted through the pipe H to the ram of the lifting

HYDRAULIC BALANCE LIFT.



Scale 1 to 96.



BALANCE CYLINDER.

Scale 1 to 32.

cylinder A. The ram rises, and in doing so causes an increasing dead weight to come into play. The pistons B and C simultaneously fall, and in doing so receive an increasing weight of water upon them, which balances the loss of head due to the rise of the lifting ram. The cage is lowered by opening the exhaust from EE (the pressure remaining, as before stated, on the top of the piston B), and as the lifting ram descends, it transmits the pressure due to its weight and that of the cage to the annular space at the bottom of the piston in the second cylinder B, and overbalances the weight of the pistons, *plus* the constant pressure on the piston B. The dead weight of the lifting ram in its descent diminishes, whilst the counter-balancing pressures on the pistons in the cylinders B and C also diminish, owing to the displacement of (and consequent reduction in the weight of) the water above them both, thus preserving the same equilibrium in the descent of the cage that was preserved in its ascent. By means of the cock F water can be admitted under pressure to the underside of the piston in the cylinder C, thus relieving the pressure at JJ in the upper cylinder, and allowing water from the top of this cylinder to flow past the packing leathers to the bottom of this cylinder, to make good any leakage or waste. By closing the cock F, whilst the cage descends, and whilst the piston C rises, a vacuum is caused, which can be utilised to raise weight in the next ascent of the lift, or to raise an empty lift without any power being exerted. The nett load raised was 8 cwt., and the water-pressure was 33½ lbs. per square inch. The speed of ascent was found to be 35 feet per minute with the cage loaded, and 138 feet per minute when it was empty. The empty cage descended at the rate of 47 feet per minute.

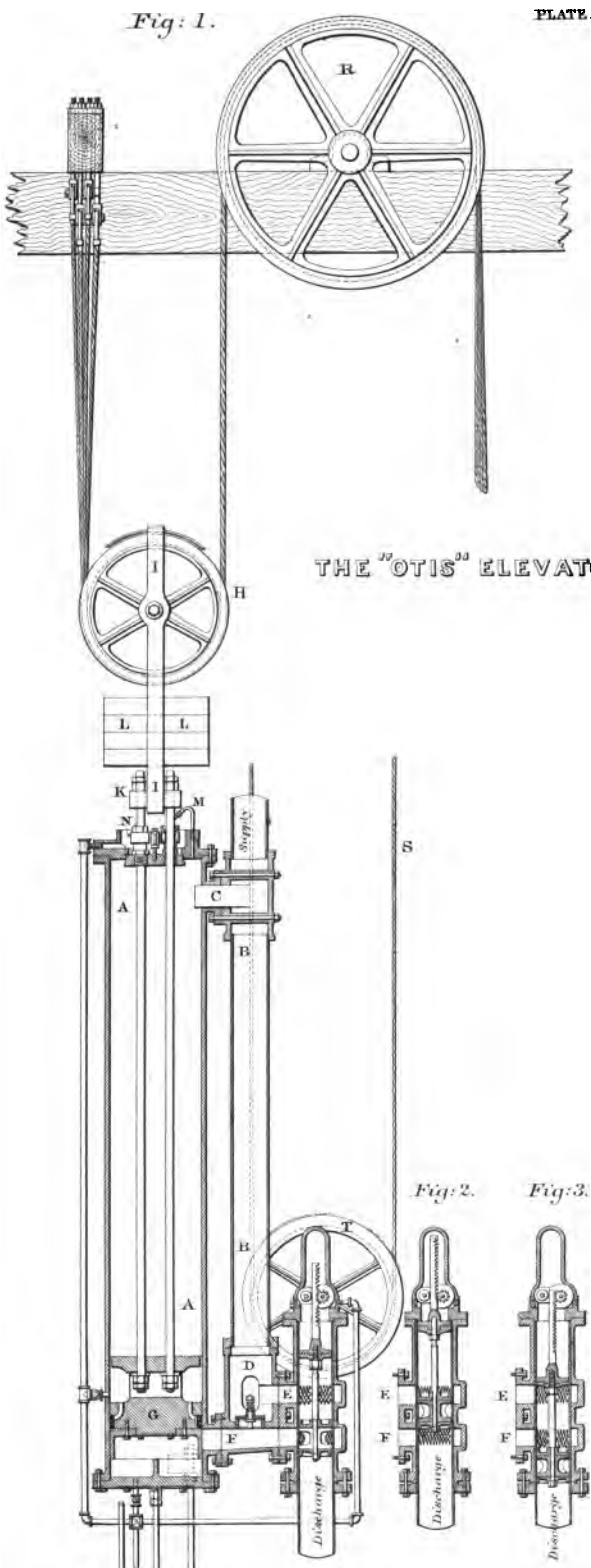
Where the lift is connected with a high pressure water main (that is, of 600 or 700 lbs. pressure per square inch), the water for the balance is proposed to be taken from and returned to a tank.

The introduction of the ingenious arrangement of the counter-

balancing cylinders is based on the assumption that entire reliance cannot be placed either on the chains of the lift, or on the safety-apparatus which comes into play when an accident happens by the breaking of the chain or wire-rope. There should, however, be no practical danger from the chain or rope (which should be connected to the sides of the cage and not the centre) if care be taken that all the materials and connections are well tested, and are under similar conditions of strain. The direct-acting lift has a greater simplicity of parts and less risk of trouble than the two hydraulic counterbalance lifts that have been described, from the various stuffing-boxes that are required in the latter.

The low pressure of water companies' mains is capable of being utilised for lifts. A good form of low pressure lift is that which is known as the "Otis Standard Hydraulic Elevator," manufactured by the American Elevator Company. The mechanical arrangements which are characteristic of this lift are shown in detail by plate 16. The motor is a cast iron vertical cylinder A connected by a tee C to a smaller cylinder B, the bottom of which rests in the water-chest D, connecting with the valve through the port E. The cylinder A is connected with the valve through the port F. The valve is a piston valve with a rack attached to the top of the piston, and is worked by the sheave T, attached to the pinion shaft, and controlled by a hand-rope S passing through the car. In the cylinder A is a piston G connected by means of two piston rods, which pass through stuffing-boxes N to a crosshead K. This crosshead rests in a double strap I, which holds the travelling sheave H, connected with the car by means of four independent wire cables, one end of each being fastened to a hitching-block by means of fork rods. The other ends, after passing under the travelling sheave H, and over the overhead sheave R, are led, two on either side, to the bottom of the car, where they are attached to the ends of the safety platform upon which the car rests.

The piston and the car thus travel in opposite directions; the former, with its attachments, balances a certain proportion of



THE "OTIS" ELEVATOR.

Thos Kell & Son, Lith.

the dead weight of the car. The rest of the dead weight is counterbalanced by cast iron blocks L placed in the strap I. Owing to the sheaves H and R the car has a travel of twice that of the piston G (the travel of which is never more than about thirty feet), so as to retain the solid column of water underneath it by atmospheric pressure, when it is at the top of the cylinder. The motive power is usually the hydrostatic pressure from the elevation of a cistern, so that the pressure rarely exceeds 40 lbs. per square inch. The speed is usually 300 to 400 feet per minute. There are three pistons in the valve connected by a stem, the upper one being for the purpose of preventing the water from escaping through the valve cap at the top of the valve. The pressure on the bottom of this upper valve piston is equally on the top of the second piston, and enables the valve to be raised or lowered without effort.

The area of the cylinder A is made proportionate to the load to be lifted. The downward pressure is always constant on the piston G, but downward motion is impossible until the column of water which is underneath the piston G is allowed to move by the opening of the valve. The exhaustion of the column underneath the piston G is effected by raising the valve piston until it occupies the space between the ports E and F as in fig. 2. This raising of the valve piston opens connection between the port F and the discharge pipe, enabling the column below the piston G to discharge, and the hydrostatic pressure on the top of the piston G to become effective in forcing the piston down to the bottom of the cylinder.

The column of water below the piston G will not fall away and discharge unless there is a pressure on the top of the piston, even if the piston G is at the top of the cylinder, as the column of water under the piston G is never more than 30 feet in height, and this column is sustained by atmospheric pressure. The available pressure is always the same throughout the entire stroke, for what is lost in head (when the piston G is near the top of the cylinder A) is balanced by the weight of

the column which hangs to the bottom of the piston; and as the piston descends, and the head increases, the weight of the column underneath the piston decreases.

For lowering the car, the valve piston is lowered below the port F into the discharge, and thus the pressure (which is also in the circulating pipe B) acts under the piston G as well as on the top of it. The pressure being thus neutralised, the car descends by gravity, raises the piston G, and displaces the column of water on the top of it. This water passes through the port C into the tee, and being prevented by the greater pressure from going up the supply pipe, it passes through the circulating pipe B into the valve, back through the port F under the piston G, filling the cylinder A under the piston, as the piston ascends. The discharge of this water is prevented by the position of the valve piston, as shown in fig. 3, and thus the water which was on the top of the piston G is led below it, ready to be discharged the next time the car is raised. The solid column of water thus acts on both sides of the piston, so that no action of the piston can take place without a displacement of water, which can only be produced by a change of the position of the valve. All motion is stopped when the valve piston covers the port F (as in fig. 1) without regard to the position of the piston G, for the column below the piston G cannot be discharged while the valve piston is covering the port F. Nor can circulation take place, because that same position of the valve piston prevents the flow of the water from the circulating pipe B under the piston G, the head in the supply preventing the water above the piston G from being forced up the supply pipe.

Attached to the piston G is a cast iron apron, or follower, which automatically cuts off the discharge at port F, in the downward stroke of the piston. The discharge is cut off before all the water is exhausted from the bottom of the cylinder A, and therefore this water forms a cushion on which the piston seats itself gradually. To prevent the accumulation of air underneath the piston G, an air valve is attached to the

piston by which the air passes through the piston to the top of the cylinder A, and then either passes out through the supply pipe or is exhausted by means of a jet cock M. When the travel of the piston G is suddenly arrested, the shock is overcome by means of a relief valve, connecting the water-chest D with the port F, enabling the water under the cylinder to communicate the shock through this valve to the column in the water-chest D, and circulating pipe B, on into the supply pipe. In case of a sudden stoppage of the piston G in an upward stroke, the shock finds vent through the port C up into the supply pipe, and is also overcome.

There are never less than four cables used, and the smallest size is half an inch diameter. The diameter decided upon in each case is such that any one cable shall have many times the necessary strength to do all the work. These cables are so attached that they receive an equal strain, and in case of the breaking of one, there is nothing to occasion the breaking of any of the others. The four cables are attached to the safety platform underneath the car, and are so arranged that the car will not work unless the strain on each cable is equal. By this means the mere stretching of one cable makes it impossible to run the car until the stretch shall have been adjusted by means of the fork rod, by which it is attached to the hitching-block.

Under the car is a safety platform, consisting of hard wood faced with iron plates. At each corner is an iron shackle rod, to each of which a cable is attached. These shackle rods are fastened to an equalising bar underneath the platform, which is held by a pivot in the centre, and so long as the strain upon the two cables is equal, the bar will retain a horizontal position, but the stretching of a cable will allow the bar to leave its horizontal position, in either one direction or another, according to which end receives the greater strain. The shape of this equalising bar is such that, when it leaves the horizontal position, the forged projections of the bar come in contact with other forgings, which are a part of the wrought iron rod that

is extended from end to end of the safety platform on its underside. One of the forgings of this rod is a finger with toothed end. The normal position of this finger is just below a brass wedge which travels with the safety platform, and is held in place by means of a shoulder both on the top and side, and is thus prevented from falling out. The platform is grooved at either end to receive the hard wood slide on which the car travels. The jaws and ends of the safety platform are faced with heavy iron plates. The position of the wedge is between the guide and one of these jaws, and, from its shape, a pressing in of the wedge creates so great an amount of friction that the car cannot travel. The wedge is pressed into its place by the finger before alluded to, and that finger is in turn worked by the mere stretching of a cable. Each end of the safety platform is equipped alike, and the rod which passes underneath the platform connects the two ends, so that action at either end necessitates the pushing in of the wedges at both ends. These wedges cannot slip out of position, nor can the slides warp out of place, or fail to be wedged. An adjustment and equalising of the tension of the cables which may have stretched, will at once remove the fingers which press in the wedges, and an upward motion of the car itself would at once release the wedges, owing to their shape. Downward motion is then possible, but it is impossible until the tension of the cables is equalised. It follows then that the heavier the weight in the car, the greater the power there is pressing in these wedges, and the teeth of the end of the finger, which comes directly in contact with each wedge.

There is also a safety governor which has a separate attachment to the car by means of an independent wire cable. This passes through the governor, under a sheave at the bottom of the well in which the car runs, and back again to the side of the car where both ends are attached. The governor is made for whatever speed may be desired, and any speed in excess of that would cause it to act.

Hydraulic power has another new field for utilisation in the

direction of working lifts for subway traffic, both vehicular and passenger. In many cases where the construction of a bridge to convey traffic over a river is objectionable, a subterranean communication has been difficult to make, owing to the approaches to the subway being impracticable. Mr. Greathead and Sir William Armstrong, Mitchell & Co., have given much attention to the question of providing hydraulic lifts, which would enable the long and expensive approaches to a subway to be dispensed with, and which would at the same time meet uninterruptedly the demands of a large vehicular traffic.

An example of this is shown by fig. 15, which represents the arrangement of hydraulic lifts proposed to be placed on Tower Hill. There are two series of cages or compartments (which are well lighted), so arranged as to admit of free ingress and egress of the traffic going in both directions. One series is for lowering the traffic going southwards through the subway, and the other is for raising the traffic coming northwards from the

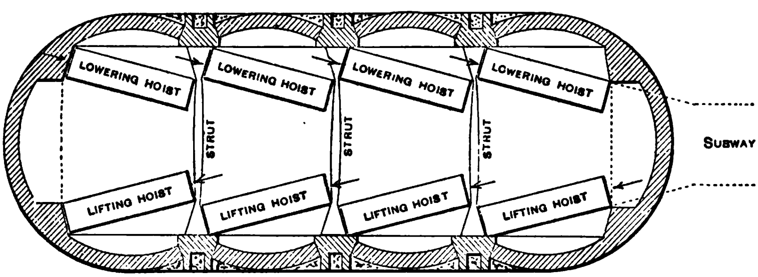


FIG. 15.

subway. Each of the compartments is of such a size as to take either the largest vehicle and four horses, or a tramway car and horses, or two smaller vehicles and their horses. The working of the lift is as follows:—A vehicle arriving would pass into, say, the first of these large compartments, and be lowered immediately to the roadway below. The vehicle following would pass into the next compartment and be lowered. By the time the last of the series of lifts or compartments had gone down, the first would be back again at the surface for a

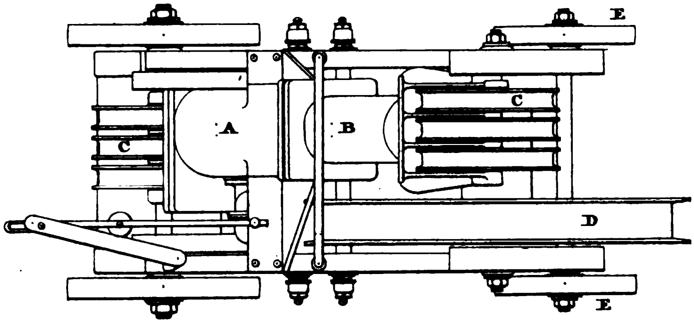
repetition of the operation. The traffic would thus pass down continuously on that side, whilst a similar series of lifts at the other end would in a similar way take it up. The cost was estimated to be £280,000.

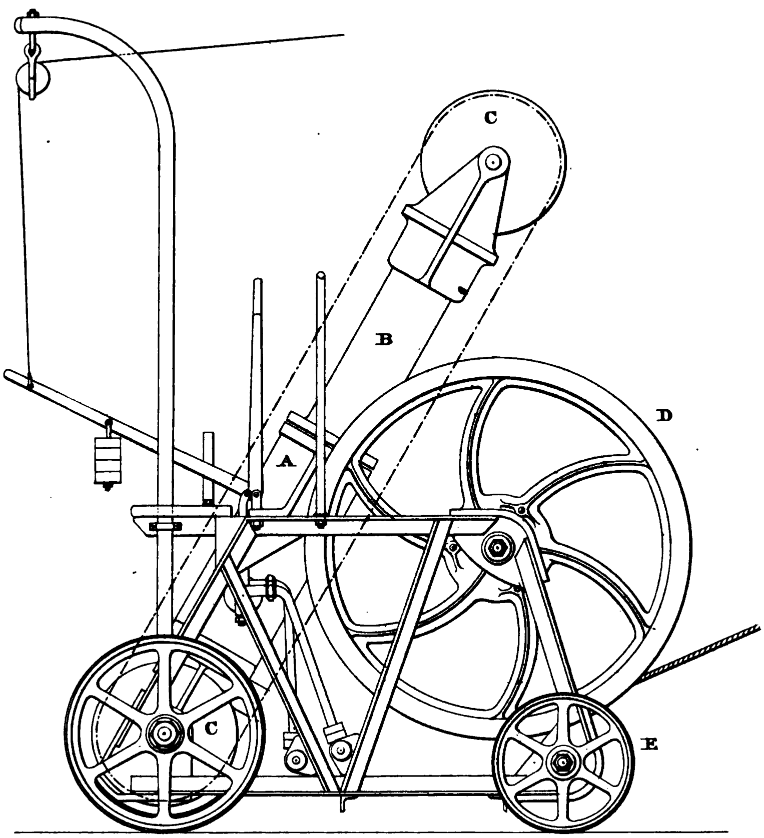
A subway can thus be approached at whatever depth it might be below the surface, and without the difficulty attending approaches with steep gradients, provided the number of lifts be proportioned to the traffic. The advantages of such a system of lifts are apparent. The continuity of traffic is not interrupted as in the case of a ferry or an opening bridge. No inclines have to be surmounted. Owing to the distribution of the traffic through a series of cages, the working expenses of lifts are in proportion to the traffic, whereas when large platforms are used, as hitherto, capable of taking a considerable volume of traffic, the working expenses are frequently out of proportion to the traffic, because at slack times the large platform is set in motion for one small vehicle. There is a great saving in first cost of communication compared with the cost of a subway having inclined approaches, or with a subway having a large platform. By the multiple-lift system struts can be put between the deep walls of the shafts (as shown in fig. 13), which would be impossible if the single-lift system were employed. An arrangement of lifts like these, effects a great saving of time to vehicles passing from bank to bank of the river. For instance, the lifts in the case shown by fig. 13 would take half a minute to go down and the same to go up, and assuming a vehicle to travel at three miles an hour through the subway it would only take five minutes from bank to bank.

The cheaper means of making communications under rivers which this system of lifts affords, appears to open out a very important extension of the application of lifts. In some cases any additional communication has of necessity to be cheaply effected. The amount, or nature, of the traffic in many cases is such as to require that only a small outlay is incurred to make the work remunerative.

MOVEABLE JIGGER HOIST,

TO LIFT 15 CWT.





Scale: 3/8 Inch = 1 Foot.

Inches 12 9 6 3 0 1 2 3 4 5 *Feet.*

Thos Kell & Son, Lith. 40, King St Covent Garden.

MOVABLE JIGGER HOIST.

A Movable Jigger Hoist is shown by plate 17. This machine consists of a hydraulic cylinder A, with a ram B, and multiplying sheaves CC. The lifting rope or chain passes over the large drum D, and the chain for communicating the power from the cylinder passes over a smaller one which is attached to it. The lifting slide valve is fixed to one side of the cylinder, and is worked by a man standing on a platform above the valve. Valve gear can be fitted to the machine (as shown in the figure) by which the jigger can be worked by a man standing on a ship's deck, and looking directly into the hold. The machine itself remains on the quay, thus dispensing with one man. It is mounted on a wrought-iron frame, carried on four wheels E, so as to allow of its being moved from place to place. The water is conveyed to the machine, from the main, through jointed pipes, which allow a considerable amount of travel of the jiggers without alteration of the pipe connections. The pressure and exhaust connections on the jiggers are shown, with caps for protecting the joints when the machine is not in use. These jiggers are of varying powers, according to the purpose to which they are to be applied, whether for lifting sacks of corn, or light jute bales. They work with great rapidity, making from four to five lifts per minute from the hold of a vessel.

HYDRAULIC WAGGON DROP.

In the arrangements for charging blast furnaces, a waggon drop is generally employed for lowering the charges into the furnace, the downward movement being controlled by a brake applied to the shaft on which are fixed the sheaves for the chains or wire ropes. Mr. Thomas Wrightson has successfully applied water as the controlling agent of the brake, and he described this arrangement at a meeting of the Iron and Steel

Institute. Fig. 16 shows the means by which water in this case is utilised as an hydraulic brake. A cylinder A (10 or 12 inches diameter) has a stroke the same as the rise or fall of the cage, which is suspended from the piston rod D, at the other end of it being the piston C, working in the cylinder. At the top of the cylinder is a small supply tank E, fitted with a self-acting ball-cock, to keep the same always supplied from the nearest water main. A small adjustable hole F in the cover communicates with the inside of the cylinder, to ensure that it is always full of water, and another small hole G in the piston allows any air which may accumulate under the piston to pass to the upper part of the cylinder, where it escapes into the tank by the hole F.

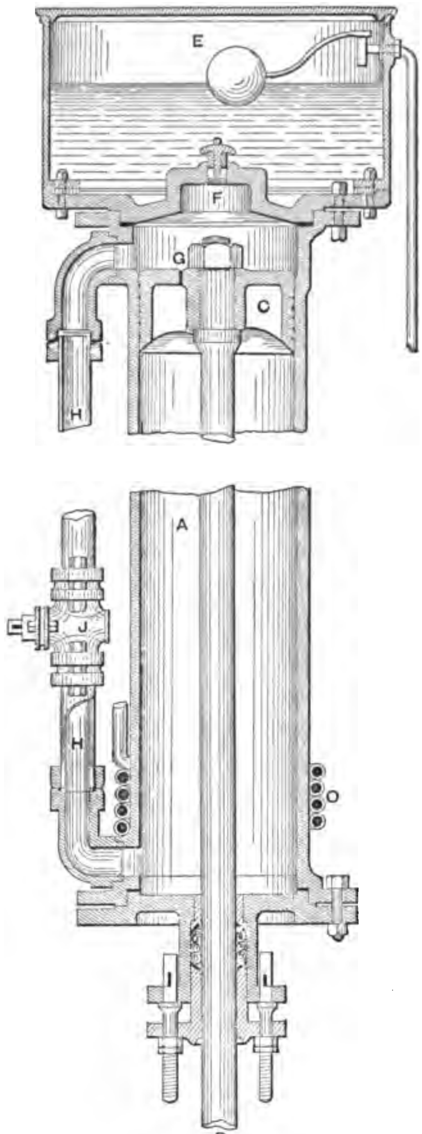


Fig. 16.

A pipe H connects the top with the bottom of the cylinder, through an ordinary water-cock J, which is controlled by a weigh-bar and lever. A catch lever is placed alongside the valve lever, and serves to lock

the cage as it comes to the top of its stroke. This holds the cage while the waggon runs on. When the cage with the waggon on is required to descend, the catch-rod is liberated, and then the valve handle is lifted. By the opening of this valve J the water passes from the bottom to the top of the piston, thus controlling the descent of the cage with the greatest nicety to any speed the attendant may choose. When the cage is at the bottom, a self-acting stop is removed by the action of the cage touching the ground, which allows the waggon to run off at the lower level. The cage being then lighter than the counterweights, is drawn up again, the water in the cylinder, during the ascent, returning from the top to the bottom of the piston. When the cage arrives at the top of its stroke, it locks itself, and is then ready for another waggon to be run on.

The bulk of the water passes and re-passes through the cock J, but on account of the area of the piston being less on the lower side than the upper (by the area of the piston-rod on the lower side), the water at the top, displaced as the piston rises, cannot find room at the lower side of the piston, and will therefore find relief by a portion (equivalent to the cubical contents of the piston-rod) passing through the small hole in the cylinder cover into the supply tank. In the same way when the piston again descends, there would be an equal deficiency in the water passing from the bottom to the top side of the piston; this is compensated for by the same amount of water re-passing through the hole in the cover. By this means the cylinder is always kept full of water, which is essential to the successful working of the apparatus. It will be observed that the same water is used over and over again, and that the ball valve in the tank is merely to supply any loss from evaporation or leakage. The small pipe O, encircling the cylinder, is for the admission of steam in frosty weather to prevent the freezing of the water. This comes from the nearest steam or exhaust pipe, and after coiling a few times round the lower part of the cylinder, passes up to the top tank alongside of the connecting pipe.

THE FLOW OF SOLIDS.

The employment, in recent years, of iron in increasingly larger masses has involved the consideration of how the continuity of the fibre can be maintained, and what the conditions are which have to be observed in order to prevent break of continuity, or a diminution of the calculated strength of the mass. The investigations of the late M. Tresca (recorded in the *Proceedings of the Institution of Mechanical Engineers,* 1867 and 1878) have thrown much light on the subject, and are of practical value in regard to forging, under a pressure or squeeze, instead of by a blow. M. Tresca applied the expression, "the flow of solids," to his investigations, and the singular facts which he established indicated that an entirely new branch of observation had been opened out, to which M. de Saint Venant has given the name of "plastico-dynamics." Fig. 17 shows the result of applying pressure to discs of lead. Ten discs of lead (each 0·12 of an inch thick, and 3·94 inches diameter) were subjected to pressure, by which the lead was forced to flow through a concentric circular orifice 1·18 inches diameter in the movable disc CC placed at the bottom of a cylinder, a plunger in which exerts the pressure.

Longitudinal Section of Jet produced from 10 Discs of Lead forced to flow through concentric circular orifice.

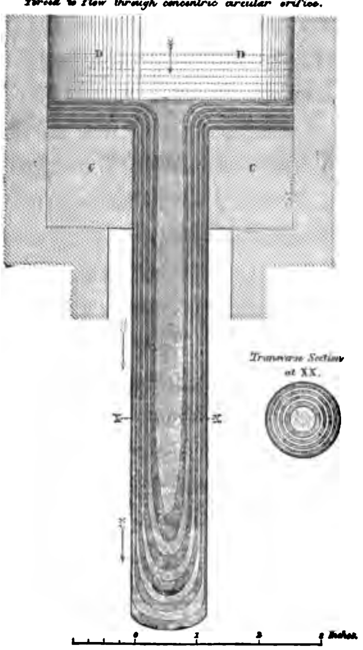


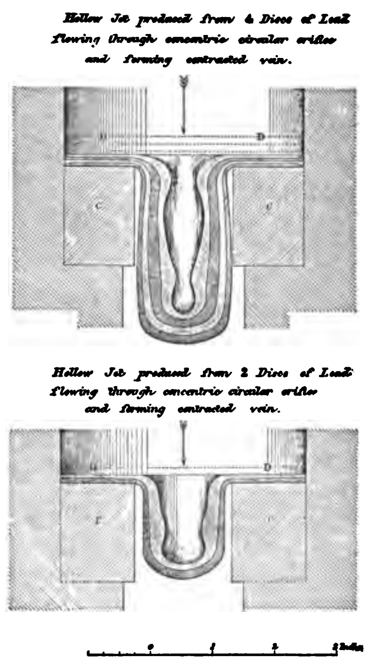
FIG. 17.

The dotted lines in the cylinder show the original positions of the discs, the upper surface being at DD. On applying pressure the jet reached 7·87 inches, which is the position in the figure. An examination of the jet proved that the layers remained flat back from the

central jet, and that they bent over from this area so as to flow into the jet simultaneously, the external surface being formed of the bottom disc, which has assumed the shape of a cylindrical covering. The other layers form separate tubes concentric with the jet, all being closed at the outer end by a cap formed of the central part of the disc.

A further experiment with a cylindrical block having a smaller height, compared with the diameter of the orifice, gave a result as shown on figs. 18 and 19.

The orifice in these two cases was 1·58 inch diameter, and each disc was 0·12 of an inch thick. DD (as before) was the original position of the layers. It is interesting to notice that the diameter of the jet is not that of the full diameter of the orifice, but a "vena contracta" has been found, such as occurs in the flow of liquids. In further experiments the undulations which were observed in the metal corresponded with the relative motions of the particles of a similar vein of fluid.



FIGS. 18 and 19.

Many other metals than lead were subject to pressure through an orifice, and the general conclusion arrived at from them was, that the particles of solid bodies flow under pressure similarly to liquids. Any alteration in the shape of the orifice, from the circular to the polygonal, or eccentric, produced torsional movements of the metal corresponding to the gyratory movements which occur in the flow of a liquid through an orifice, which is not placed symmetrically to the sides of the vessel containing it.

When the metal was pressed through more than one orifice in the die, it was observed that the jets nearest the centre were rather larger than those near to the sides of the cylinder, the lesser effect being due to the friction of the sides. This difference in pressure on different parts of a solid mass explains the displacements that take place in the interior of the mass.

The experiments established that the pressures exerted on the surface of a solid body are transmitted throughout the whole interior of its mass, and tend to produce in it a flow which is propagated from particle to particle, and which necessarily develops itself in the direction where the resistances to the flow are the least; also that the pressures thus transmitted determine in a fixed order the changes of form at each point. Further, these changes of form are attended by a loss of pressure between one point and another, similar to, but even greater than, that in the case of the flow of liquids.

In the processes of rolling and forging iron, the observations of M. Tresca have a practical value, as indicating the necessity for the application of a pressure or blow sufficiently powerful to reach the interior of the mass in order to enable the metal to flow, and its fibrous continuity to be preserved.

The object which has to be attained in manipulating wrought iron under a forging, bending, or other tool, is to dispose the fibres in the direction conforming to the purpose to which the iron is to be applied. Such disposition of the fibres or threads in uniformly continuous lines ensures the strength of the mass being preserved, whereas if the application of a blow results in a disturbance of this arrangement of the fibres (producing, as it were, eddies in the flow of the particles), the power of resistance is necessarily lessened.

The application of a squeeze by a hydraulic appliance, instead of a blow by a steam-hammer, enables the cperation of stamping metal to be performed with better regard to the preservation of the continuity of the particles. The shapes for the dies at the various stages of the work can be considered with reference to the natural tendency of the metal to flow in the direction of the

pressure which is applied to it. The element of time in these operations has been proved incontestably to be an important factor in the changes of form which are produced in metals when being manipulated. That is to say, the continuity in the fibres of the metal is better produced by a slow blow or squeeze, than when the operation is performed more rapidly, as is the case with a sharp blow.

The application of hydraulic power at high pressures to actuate machines to produce the results thus indicated, has opened out a wide field for the extension of its employment. The important part that hydraulic power will play in the future manipulation of iron and steel justifies a somewhat lengthy reference being made to this branch of the subject.

The employment of hydraulic pressure to workshop tools dates as far back as Bramah's time, a hydraulic planing machine having been then erected at Woolwich, in which many of the operations of the tool were performed by water-pressure. More recently (about thirty years ago) a direct-acting hydraulic slotting machine was at work for some years at Elswick. It had a stroke of about 4 feet, with a tumbling weight to reverse the action, and it was worked with an accumulator pressure of 700 lbs. per square inch. It was placed vertically upon girders supported by pillars 16 feet apart, thus enabling large pieces of machinery to be easily slotted.

Hydraulic power is now generally applied at high pressure to actuate shop tools. The installations that are now laid out to work an extensive series of appliances, may be regarded as complete systems of producing, storing, and distributing water-power at the high pressure of 1500 lbs. in the same way that Sir William Armstrong perfected the original system of hydraulic power distribution, and utilisation, for the multifarious purposes to which it has been, and still is, applied at the lower pressure of 600 or 700 lbs. to the square inch. In both cases the machines to which the power is most advantageously applied are direct acting, and the work to be performed is intermittent.

The adoption of a pressure of 1500 lbs. per square inch

enables the sizes of shop tools to be reduced, and their portability and convenience thereby increased; but in case a lighter description of work has to be done, the pressure can be reduced by diminishing the weight in the accumulator.

The transmission of power by a pipe (instead of by belting or shafting) for actuating shop tools, is attended with advantages. Less wear and tear arises, and the power can be conveyed round bends, or to distant points, with great facility. The pipes being underground, the cost of the supports, columns, and bearings requisite for shafting is saved.

In the application of water-power to actuate machines there is a great advantage in employing it at high pressure. The power (or foot pounds) transmitted through a high pressure water main is determined by multiplying the number of pounds of water flowing per second by the pressure. If the same number of pounds of water are delivered at 1500 lbs. pressure to the square inch, double the energy is exerted as would be the case with a pressure of 750 lbs. per square inch. If the same quantity of water is pumped through two pipes of the same diameter, the water in one being pumped at 750 lbs. per square inch, and in the other at 1500 lbs. per square inch, then every pound of water at 750 lbs. pressure contains 1700 foot pounds of energy, and every pound of water at 1500 lbs. per square inch contains 3400 foot pounds of energy. In transmitting this volume of high pressure water, whatever number of foot pounds of energy were lost by friction in the water at the lower pressure, no more would be lost by the water at the higher pressure, as friction is independent of pressure. But as only half the high pressure water will be required to do the work in comparison with the lower pressure water, the necessary velocity of the high pressure water would be half that of the low pressure, and consequently the loss by friction would be less.

To Mr. Ralph Hart Tweddell is mainly due the successful solution of the many practical difficulties which had to be overcome in applying hydraulic power to tools. In 1865 he devised a small portable hydraulic apparatus for fixing the ends of boiler

tubes in tube plates. These ends have to be made fast by enlarging them at their junction with the tube plate, and by hammering over the end, or by driving ferrules to hold the tubes tightly in position. By the hydraulic method referred to, the operation is effected by means of a disc (formed in sections) in the end of the tube. This disc is pressed outwards, and the enlargement is produced by drawing a conical rod through the centre. Hydraulic pressure acting upon a piston attached to the tapered rod effects this. This tool is mentioned on account of the mechanical ingenuity displayed in utilising water-power for the purpose for which it was invented. Its practical utility has, however, been found to be small, owing to the care that had to be taken in working it.

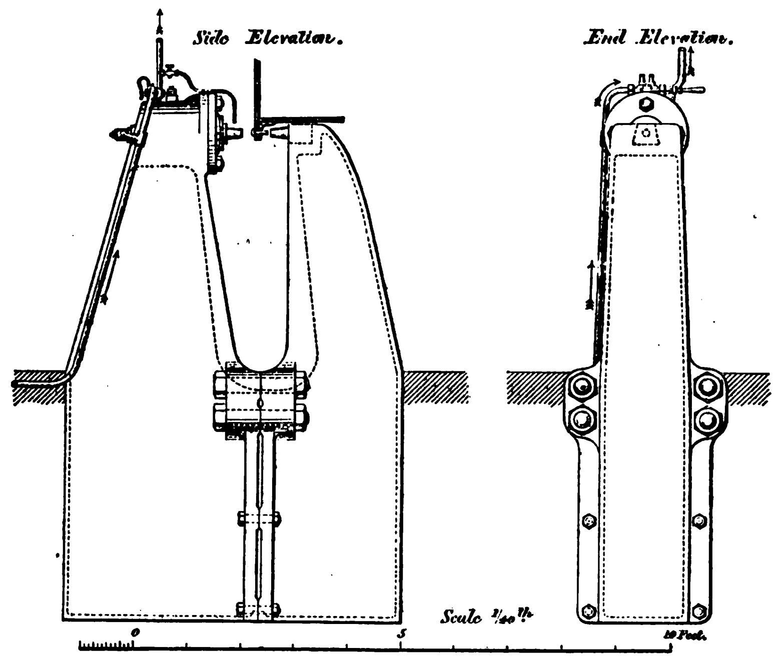
Hydraulic rivetting was carried out at the Elswick Works as far back as the year 1851, and the machine is still working. It was originally a steam rivetter, but as its action was not satisfactory, Sir William Armstrong converted it into a hydraulic rivetter, by simply altering the piston and valves, and by applying a set of force-pumps, connected with an accumulator which could be loaded to various pressures to suit the strain required to be given upon the varying sized rivets, the largest being one inch. The maximum accumulator pressure that is applied to this machine is 300 lbs. to the square inch, which cannot be exceeded, owing to the water-pressure being applied to the original steam cylinder.

When Mr. Tweddell directed his attention to hydraulic rivetting at a later date, he first experimented with steam rivetters, and diagrams were taken to determine the best pressure to produce a good rivet. This was especially directed to rivets in the thicker plates (such as 1¼ inches) that came to be adopted a few years ago for large boilers, and other descriptions of plate-work, where 1⅜ inch rivets or so were employed. A hydraulic rivetting machine was then designed by Mr. Tweddell, which proved that only a small amount of friction occurred in the machine. The advantage of the momentum due to the descent of the accumulator, when water was passing from

it into the cylinder of the rivetter, combined with the steady pressure or squeeze due to the load, was established.

One of Mr. Tweddell's fixed hydraulic rivetters is shown by figs. 20 to 27. A side elevation is given by fig. 20. An end elevation by fig. 21. Fig. 22 is a plan and fig. 23 a longitu-

FIG. 20. FIG. 21.



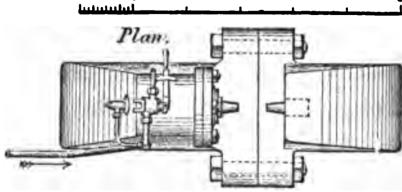


FIG. 22.

dinal section. Fig. 24 is a back elevation, and fig. 25 is a front elevation. Fig. 26 is a sectional plan of the valve box. Fig. 27 is a longitudinal section of the ram. These illustrations are taken from the *Proceedings of the Institution of Mechanical Engineers.*

The high-pressure water is conveyed to the machine by lengths of copper pipes twisted spirally, and having universal joints, which arrangement forms a good elastic connection, capable of

FIG. 23.

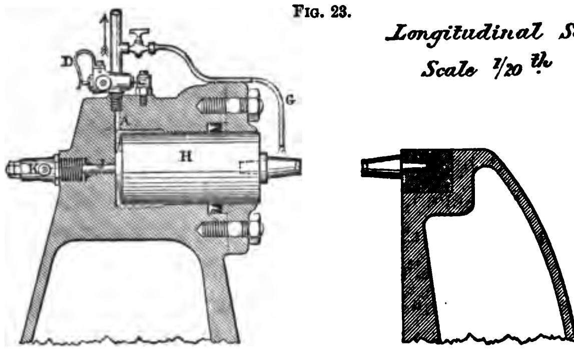


FIG. 24.

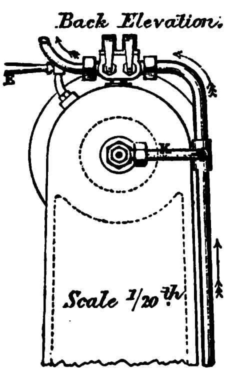


FIG. 26.

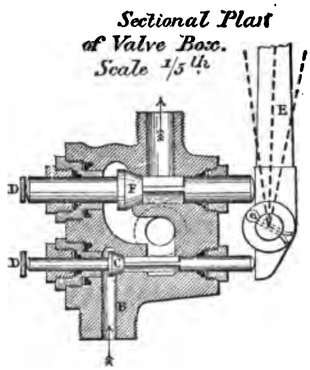
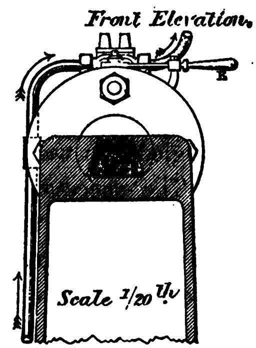


FIG. 25.



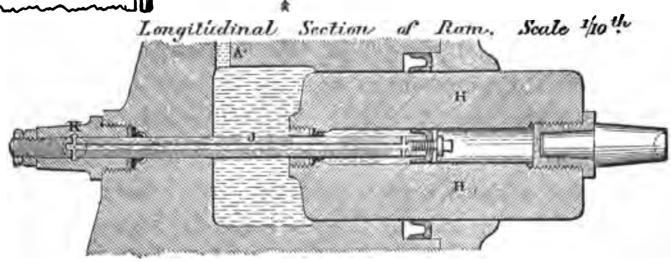


FIG. 27.

being turned in any direction. The water from the accumulator is admitted to, and exhausted from, the cylinder through the

small aperture A., fig. 23, by means of a simple hydraulic valve shown by fig. 26. The water entering at B tends to keep the inlet valve C shut, the spring D serving the same purpose until the accumulator pressure begins to act. On opening the valve C (by the hand lever E) water is admitted to the cylinder and passes into it against the ram (8 inches diameter), until the rivet is closed. The exhaust valve F is kept shut by the pressure of water entering the cylinder, and at other times by the spring D, but by pulling the lever over the reverse way the exhaust valve is opened, by which the exhaust water escapes to the cistern, a small portion being allowed to flow through the pipe G (fig. 23) on to the die, to cool it. The ram H is drawn back by means of the small drawback cylinder J (fig. 27), which is arranged within the ram itself, and is in constant communication with the accumulator through the inlet K. The wedge-shaped fastening of the disc (as shown at L in figs. 23 and 25) obviates any thickness of metal over the fixing pin ordinarily employed to keep the die in its place. This enables the rivetter to be used for rivetting flanged and angled iron work.

The success which resulted from the use of water-power when applied to the fixed rivetters, led to its application to a portable rivetter, which was able to be taken to the work, instead of the work having to be brought to the rivetter.

Figs. 28 and 29 show a sectional elevation and end elevation of one of the portable hydraulic rivetting or punching machines. In the cylinder A is a plunger or ram B, with two jaws CC, to which is attached a cross-head or horn D, furnished at one of its ends with the cupping die E. The plunger B, being forced forward, by water admitted through a valve by the gearing F, the cross-head advances until it meets the resistance of one end of the other cross-head G at a point H, which is shaped to receive J, and at the other it comes in contact with, and closes, the rivet, or punches or shears the plate, according to the purpose it is used for. At the same time the outside cross-head (or one farthest from the cylinder) is held up against the

cross-head D by two tension rods KK attached to the cross-head G at LL, and to lugs cast on the cylinder, the rods receiving the thrust. The horns or cross-heads are supported by the through pins MM, whilst the horn which is attached to the plunger B is steadied by the guide N. The nuts OO regulate the distance from the face of the cylinder A to the centre line of the cross-head G. The ram B is always subject to a drawback action owing to the self-acting cylinder P having the

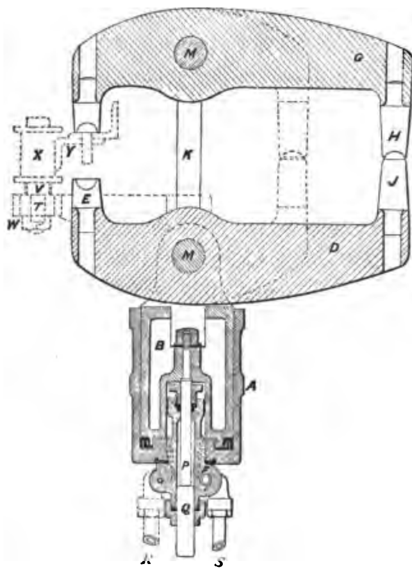


FIG. 28.

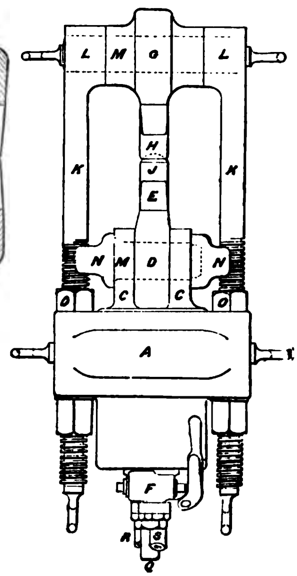


FIG. 29.

accumulator pressure constantly exerted upon the shoulder Q. This power comes into action as soon as the water is exhausted through the valve F. The water-pressure is admitted to the cylinder P by the inlet pipe R, and is exhausted by the pipe S, both being worked by a handle.

The dotted lines show the arrangement for keeping the machine up to its work. A frame T is fixed to the outer horn D, and in this is a pin V attached to the frame by a

nut W, and having on it a roller X, which has a free rotary motion. The roller is covered with some flexible material, and serves to keep the machine in its right place, and up to the work Y.

A form of portable hydraulic rivetter is made which can be fixed to a bracket temporarily at any part of a yard where rivetting has to be done, that admits of the temporary setting up of a portable machine.

Where plate-work has to be put together abroad, the conditions of labour render it important to ensure thoroughness of work in rivetting, and it is often worth while to set up a small hydraulic power installation to rivet by these machines, so that the certainty of good work, with the minimum of hand labour, is ensured. Where the transport of the heavy weights for ordinary accumulators is a difficulty, a high pressure on water can be obtained by applying low pressure water from a cistern to the large (or low pressure) piston of an "intensifying accumulator" already described. By means of a portable engine, water is pumped into the small end of this accumulator, from which it is conveyed by a high-pressure pipe to the machines. A pressure of 20 lbs. per inch can be obtained on the large end of the piston of the intensifying accumulator with a head of water in a cistern of about 40 feet, and as this water is not consumed, but remains permanently acting on the larger area, no waste takes place. A small tank holding 30 gallons will keep a 6-inch portable rivetter in full work, at a pressure of 1500 lbs. per square inch.

Hydraulic rivetters are applied with great advantage in shipbuilding, and the results lead to the opinion that the strength of the ships which are thus rivetted is increased. The best authorities are agreed that on machine rivetting the strength of the steamships of the future, with their increasing engine-power, and with the corresponding strains and vibrations, largely depends. By means of hydraulic rivetters, not only is the quality of the work improved, but a labour-saving appliance is employed in a class of work requiring but little skill.

At the London & North Western Railway Locomotive Works at Crewe, Mr. Webb employs hydraulic rivetting machines to put in all the rivets that he can in the locomotive side frames. Some years ago he rivetted up one or two pairs of locomotive cylinders with an hydraulic machine, and these have been at work ever since. He considers that work of this kind done by hydraulic machines is superior to, and cheaper than, work done by hand or steam. He examined some plate work that had been rivetted under a pressure of 47 tons, and it was found that the plates had not suffered at all by the action of the hydraulic rivetter. The "drifting" of the holes, which is necessary in hand rivetting, he considers to be more productive of injury to the plates than any combined squeeze and blow of a hydraulic rivetter. The closeness of the work that hydraulic rivetters turn out has been proved by the fact that some portable boilers made by them have been found to be steam-tight without caulking. The plates in these cases were $\frac{3}{8}$ths of an inch thick, and the rivets $\frac{5}{8}$ths of an inch. The avoidance of caulking is important, as it prevents the interference with the close contact of the plates, which occurs sometimes in caulking.

The application to a rivet of a combined blow and squeeze (such as is obtained from a hydraulic rivetter) prevents the formation of a shoulder on the rivet between the plates, such as is produced occasionally with machine rivetting, when a sharp powerful blow is applied. It is desirable to prevent shoulders, as they involve drilling out the rivet, and the caulking of the joint.

The action of hydraulic pressure in rivetting operations has been well shown by indicator diagrams taken from the pressure cylinders of the Tweddell rivetting machines at the Toulon Dockyard. Professor Unwin pointed out (in a lecture on Water Motors at the Institution of Civil Engineers) some interesting features which were exhibited by these diagrams, as they differ altogether from those taken from an ordinary steam-engine. A steam-engine is actuated by a fluid of comparatively little

weight. Water, however, being 500 times as heavy as steam, involves the consideration that its weight acts with, and increases that of, the piston. For instance, in the case of a rivetter worked from a differential accumulator through a 1-inch pipe, the velocity with which the water is forced by the accumulator through the pipe to the rivetter is increased or diminished according to the speed of the rivetter ram, by which the mass of water in the accumulator cylinder and pipe acts both to increase or diminish the effect produced by the ram. Assuming, as is the case in practice, that the motion of the loaded ram of the accumulator is six times as fast as the rivetter

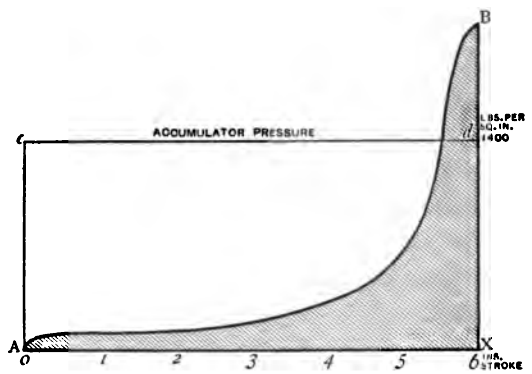


FIG. 30.

ram, the inertia of the accumulator load is 36 times as great as it would be if it moved at the same speed as the rivetter ram. Further, the force due to the inertia of the water passing into the rivetting cylinder is more than 6000 times as great as would be the case if the water and rivetting ram travelled at the same speed, owing to the fact that the velocity of the water is 81 times as great as that of the ram. This results in the weight of the ram, which closes the rivet at each stroke, exerting a force of 300 tons.

Professor Unwin has shown by indicator diagrams the action that takes place in a rivetter cylinder. Fig. 30 is a diagram

from a rivetter driven by a differential accumulator through 30 feet of 1-inch pipe. It will be seen that whilst the pressure is least at the beginning of the stroke, it jumps up at the end of the stroke above the accumulator pressure of 1400 lbs. per square inch, showing the action of the machine to be favourable to the work to be performed, by slowly closing the rivet at first, and then by bringing the maximum pressure, in the form of a squeeze, at the last. The rectangle A c d X would be the diagram without friction and inertia, but the actual pressure is much less, being only the shaded part of the figure. Fig. 30A gives an analysis of the friction.

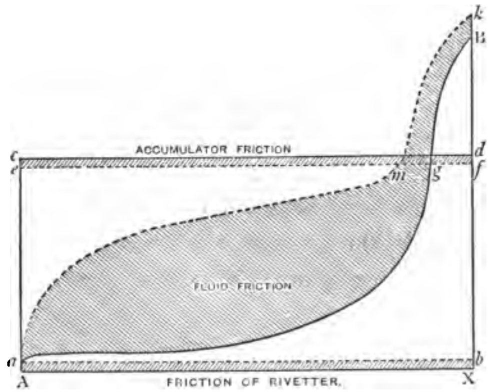


FIG. 30A.

The friction of the cup-leather of the rivetter is shown by the small shaded rectangle A a b X, and the friction of the packing of the accumulator is shown by the small shaded rectangle e c d f. The friction of the water in the 1-inch pipe is shown by the large shaded surface a m k B g, and this friction maintains the safe working of the machine at about a foot per second. The two blank spaces a e m and m k f represent the stored work in the first half of the stroke, and the excess of work at the end of the stroke, respectively, which is also shown by fig. 30B.

Rapidity and economy result from the use of hydraulic power

in rivetting, as compared with hand labour. Even in heavy work, hydraulic rivetters have put in 1000 1⅛-inch rivets in 1-inch plates in an ordinary day's work of 10 hours. In portable boiler-work the average rate of working is 7 rivets per minute, and it has been recorded of one machine that it put in an average of 5000 rivets a day for several weeks. With adequate accumulator power 15 rivets per minute can be put in.

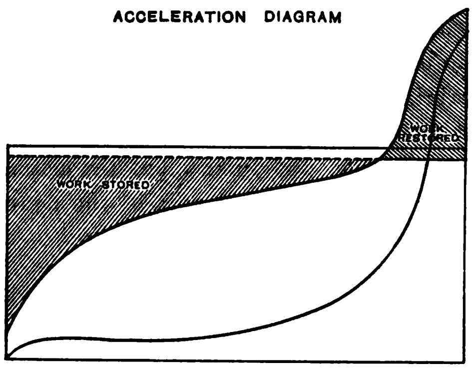


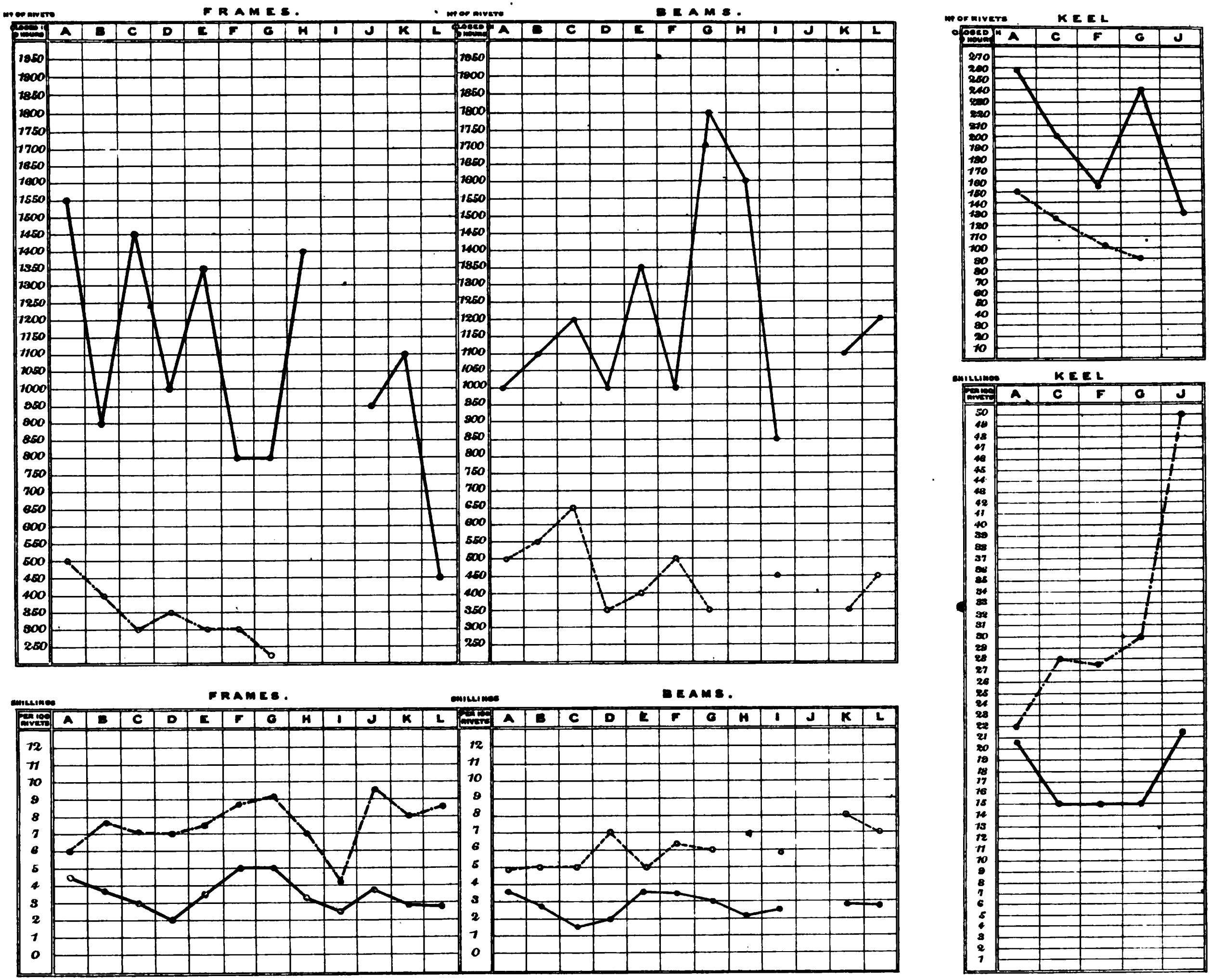
FIG. 30B.

The annexed diagram gives the result of a large number of observations.

The quality, economy, and superiority of the work performed by hydraulic rivetters indicate that it will be universally adopted in the future where water-power is available, or where the permanency of the demand for the power justifies the installation of it. Many interesting experiments have been made by Professor Kennedy on "Rivetted Joints," the results of which have been described in papers to the Institution of Mechanical Engineers. In considering the strength of a joint it is important to notice not only the strain at which fracture takes place, but also that at which the joint begins to give way by slipping. Judged by this standard the machine work was shown to be much stronger than the hand work. In hand rivetting ⅝-inch plates, it was found that slipping began at 27 per cent. of the

PLATE. 17 A.

RELATIVE RATES OF MACHINE & HAND RIVETTING.



RELATIVE COST OF MACHINE & HAND RIVETTING.

Thoˢ Kell & Son, Lith.

breaking strain, whereas in the machine joints the slipping did not commence till the strain had reached 59 per cent. of the breaking strain. In the hand-rivetted ¾-inch plates, slipping began at 16 per cent. of the breaking strain, and in the machine-rivetted at 28 per cent. This appears to show conclusively that, regarded from a practical point of view, machine rivetting possesses very decided superiority over hand work. In these experiments Tweddell's machines were used, and the pressure upon the rivet-heads was 35 tons per square inch. The loads per rivet at which slipping began were found to be for ¾-inch

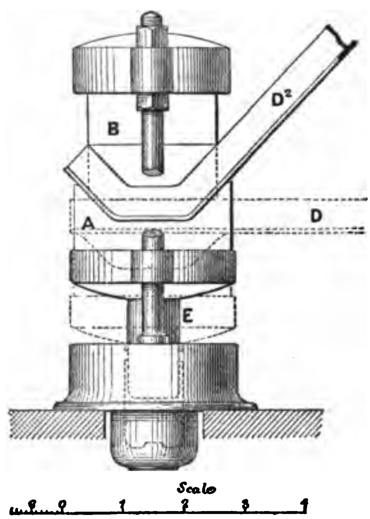


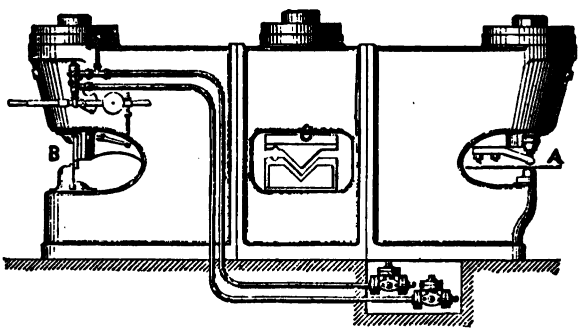
FIG. 31.

rivets single rivetted by hand 2½ tons, for ¾-inch rivets double rivetted by hand 3 to 3½ tons, for ¾-inch rivets double rivetted by machine 7 tons. The corresponding loads for 1-inch rivets were 3·2, 4·3, and 8 to 10 tons respectively. It is thought that the load at which visible slip commences is probably proportional to the load at which leakage would occur in a boiler.

The nature of the work that is performed in punching holes in plates, and in shearing or bending plates, points to the application of water-pressure to obtain the direct action required for those operations. A machine on this principle, called a "Hydraulic Joggling Press," is shown by fig. 31.

Suitable dies A and B are fixed in the hydraulic press. The dotted line shows the bottom table in position, when the angle of the T bar D to be bent is placed on the top of the die A. On the ram E rising, the angle iron is bent between the two blocks into the required form at D^2.

Figs. 1 and 2, plate 18, show one of Tweddell's machines for shearing chain cables. It will be observed that the knife A is



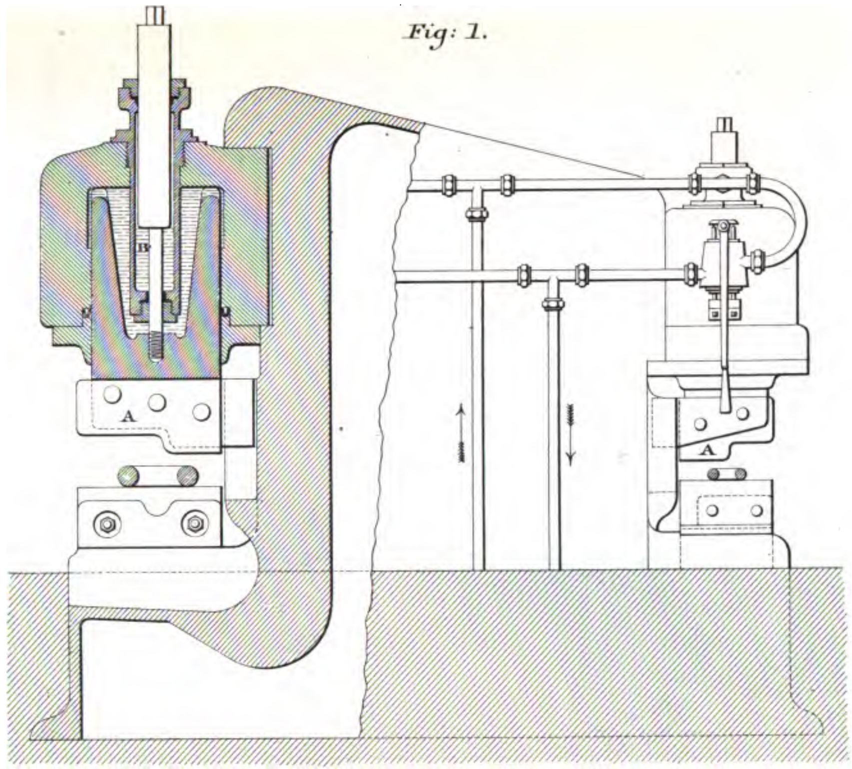
3/16 = 1 Foot

FIG. 32.

stepped, so that although the cable is cut by one stroke of the machine, it is not done at the same moment. This enables the cylinder to be reduced to one-half the size that would be required were both sides of the cable to be cut simultaneously; and as the pressure employed is usually 2000 lbs. per square inch, the reduction of diameter that can be effected is a great consideration, as a strong cylinder can be obtained with a moderate quantity of metal. The drawback gear B is also worked by hydraulic power. The machines have two ends, so that any size of cable from 3½ inches to ½ an inch can be

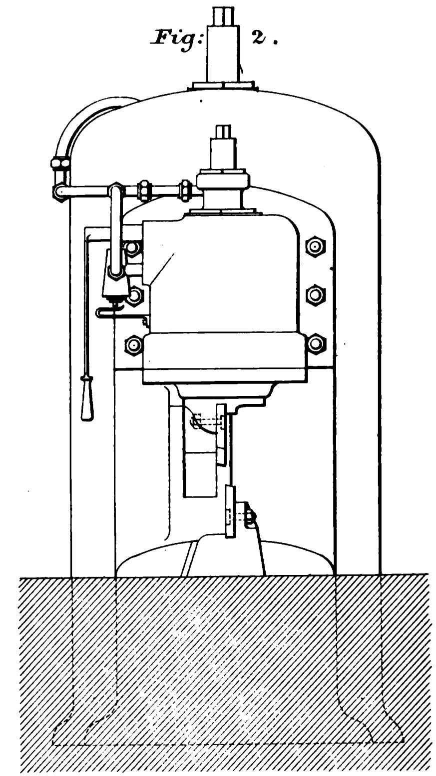
HYDRAULIC CHAIN-CABLE SHEARS.

Fig: 1.



SIDE ELEVATION.

Fig: 2.



END ELEVATION.

Scale 1/20th

Inches 12 6 0 1 2 3 4 5 Feet

Thos Kell & Son, Lith.

cut without changing the knives, or injuring an unnecessary number of adjoining links.

Fig. 32 shows a general elevation of a Tweddell triple punching and shearing machine, capable of punching $1\frac{1}{4}$-inch holes in $1\frac{1}{4}$-inch plates, or shearing $1\frac{1}{4}$-inch plates, or angle-irons $6\frac{1}{2}$ inches by $6\frac{1}{2}$ inches by $\frac{5}{8}$ inch. The several tools for performing these different operations are shown on fig. 32 at A, B, and C. Fig. 33 is partly an elevation and partly a

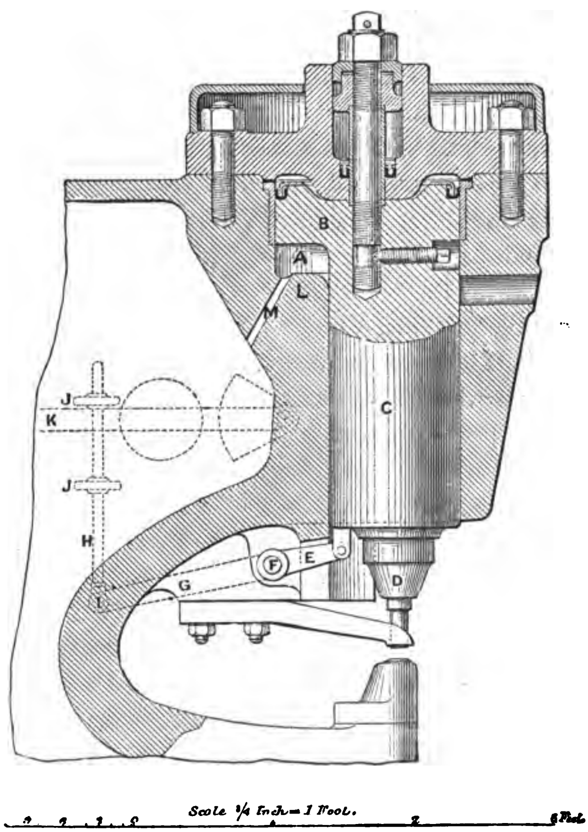


FIG. 33.

vertical section of the punching-machine. A is the cylinder; B is the piston, which is prolonged by an eccentric stem C, to which is attached the punch-holder D. The stem C, being

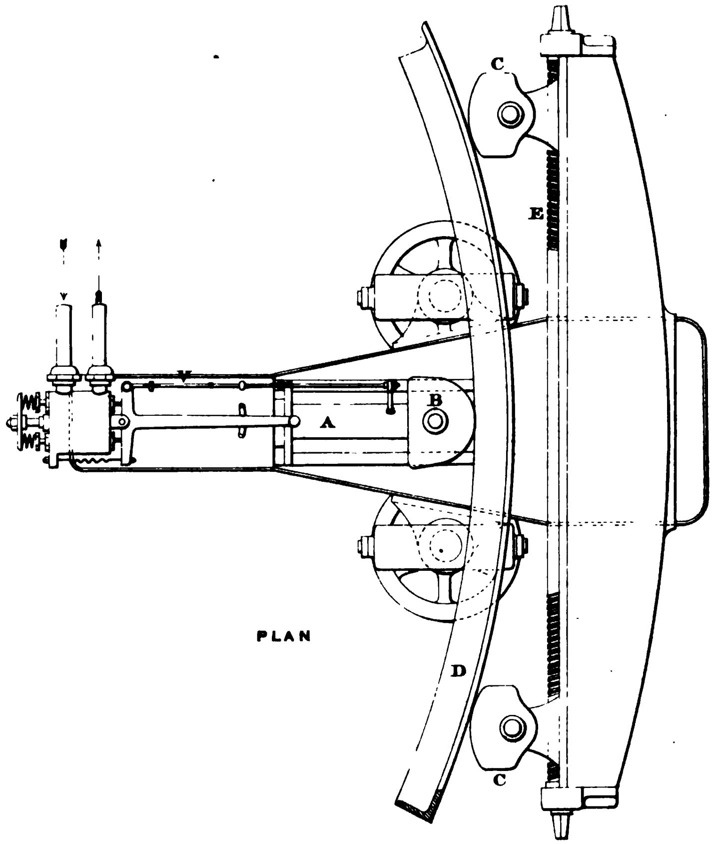
eccentric to the piston B, maintains the moving tool constantly in its right position relatively to the fixed tool, by preventing any twisting action going on while it is at work. The shoulder L forms a stop. The piston is raised by admitting water through the hole M. E is a lever moved by the stem C, and it actuates the rocking-shaft F, upon the other end of which is a lever G connected by a flexible joint I to the rod H. By means of the adjusting nuts JJ (which strike the valve-lever K) the stroke of the piston can be regulated according to the thickness of the plate that is being punched. The shearing tools shown at B and C are actuated in a similar manner.

These machines are made of great power to enable dies or punches to be attached for punching large openings in thin plates, and for stamping and moulding. All the dies and knife-holders are made so as to be readily detached, and the necessary moulds applied for cutting and stamping. The knives can be placed at any angle.

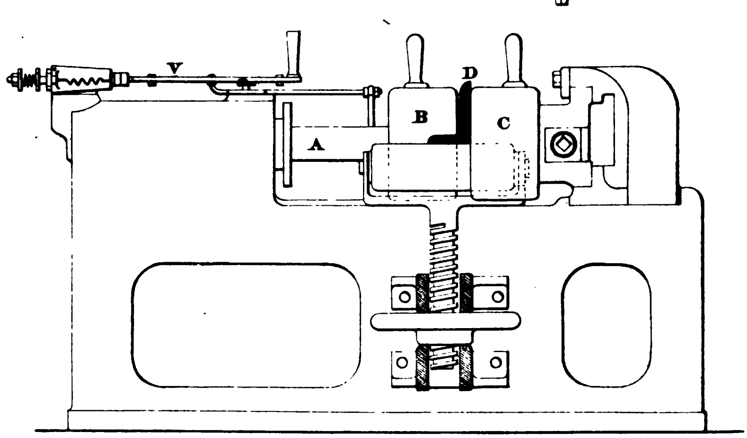
A shearing machine, the ram of which has an effective diameter of 14·05 inches and 7·87 inches stroke, requires 1354·75 cubic inches of water to be expended from the accumulator per stroke, and transmits a pressure of 98·42 tons. The maximum work done by the machine per stroke is 144,660 foot pounds, while the corresponding power developed by the prime mover is 216,990 foot pounds. In a working day of ten hours the ram would deliver 250 strokes the volume of water expended would be 196·21 cubic feet and the work expended would be 54,247,500 foot pounds.

Probably the most complete installation of hydraulic shop tools is that which the French Government have introduced at the iron shipbuilding department at Toulon Naval Dockyard, under the advice of M. Marc Berrier-Fontaine, Ingéniéur to the Marine Toulon Dockyard. A detailed description of this installation was given to the Institution of Mechanical Engineers. The machines are supplied with water at a pressure of 1500 lbs. per square inch by means of cast-iron pipes 2½ inches internal diameter, which were tested to double that pressure. The tools

5 TON ANGLE-IRON BENDING MACHINE.



PLAN



SIDE ELEVATION.

Scale. 1/20th

Ins. 12 9 6 3 0 1 2 3 4 5 Feet

Thos Kell & Son Lith

are placed on one side of a shop, with gas furnaces on the other, for testing the various plates, angle, and channel irons of iron-clads.

The machine shown in plan and side elevation on Plate 19 is for angle iron bending at the Toulon Dockyard. The ram A forces the block B against the iron D to be bent, and the positions of the blocks CC (adjustable by the screw-spindle E) determine the curve to be formed. The vertical-screwed rod V has its upper end pinned to the starting lever, and its lower end passes through an eye in the extremity of a horizontal arm carried by the ram of the machine. On the screw are two nuts, one above and the other below the arm, which serve the following purpose:—On the ram rising, the arm strikes the upper nut, and acts on the standing lever with greater force than a man could exert. The exhaust is thus closed, and the upstroke of the ram is arrested. When the ram descends, the arm strikes the lower nut on the screw, which closes the starting valve, and stops the ram in its descent. By adjusting the positions of these nuts, the length of the upward or downward stroke may be regulated to suit the work to be done.

Many years ago, Mr. E. A. Cowper employed a hydraulic reservoir, in conjunction with a press, at the works of Sir Charles Fox, to squeeze wrought iron into shape. Some heavy links for the Kief Suspension Bridge were made by means of this press. They were 7 feet 6 inches long, 1 foot 4 inches wide, and 1 inch thick, the eyes and long slots being cut out by the press.

In 1861, Mr. John Haswell employed a hydraulic press of 800 tons for forging parts of locomotives in cast iron dies, and to him is due the introduction of a press on the Bramah principle, with the addition of a motor, by which a squeeze can be given to the metal.

In 1862, Mr. James Tangye described (in a paper read before the Institution of Mechanical Engineers) a simple application of the hydraulic press to shearing and punching. For the former operation, a press carrying a knife edge was connected

by bolts to a frame carrying another knife edge. The piece of metal to be sheared was placed between the two edges, and the press being set in motion by means of a small hand-pump, the knife was forced through it. A bar 3 inches square was cut through in about 2½ minutes. The shears were well adapted for cutting rails. The action of the punch was of a similar character, and one man could punch an inch hole in a ⅞-inch iron plate in about half a minute. These machines were light and portable, the shears weighing about 14 cwt., the punch 4½ cwt. Mr. Tangye, at the same time, described a hydraulic jack for lifting weights up to 60 tons. Hydraulic power was also employed at his works for stamping purposes, an air vessel being employed to produce an intensification of pressure.

In 1869, Mr. Moss of Chicago applied hydraulic pressure for compressing steel, from which wheels for locomotives and trucks were made.

In 1864, Sir J. Whitworth erected a hydraulic press to forge and to press fluid steel, and the result showed such advantages over hammer forging, that another was erected in 1870, with a 24-inch cylinder working at a pressure of 2 tons to the square inch. Two others having 30 and 34-inch cylinders followed, with a pressure of 3 tons to the square inch, although the working pressure rarely exceeded 2 tons to the square inch. The forging presses are all worked by single cylinders in the press heads In that for pressing fluid steel there are three cylinders below the base plate, which is raised by them. On it is a mould containing fluid steel which is pressed against a fixed plunger in the press head. By the application of intense pressure to fluid metal in a mould, its whole length can be diminished one-eighth in less than five minutes, the air cells being expelled. The two screw propeller shafts of H.M.S. *Inflexible* were made from pressed metal a few years ago at the Whitworth works. They were 283 feet in length, 17 inches in diameter, with a 9-inch hole through them. They weighed 63 tons, compared with 97 tons, which it was estimated would have been the weight of wrought iron shafts. The strength of the compressed metal

shafts was 40 tons to the square inch, and the ductility was 30 per cent. The pressure applied was 8000 tons, and it was found necessary to employ a pressure of from 6 to 9 tons per square inch as soon as the metal was run into the mould, as the gases could not be expelled when the metal was in a semi-fluid state.

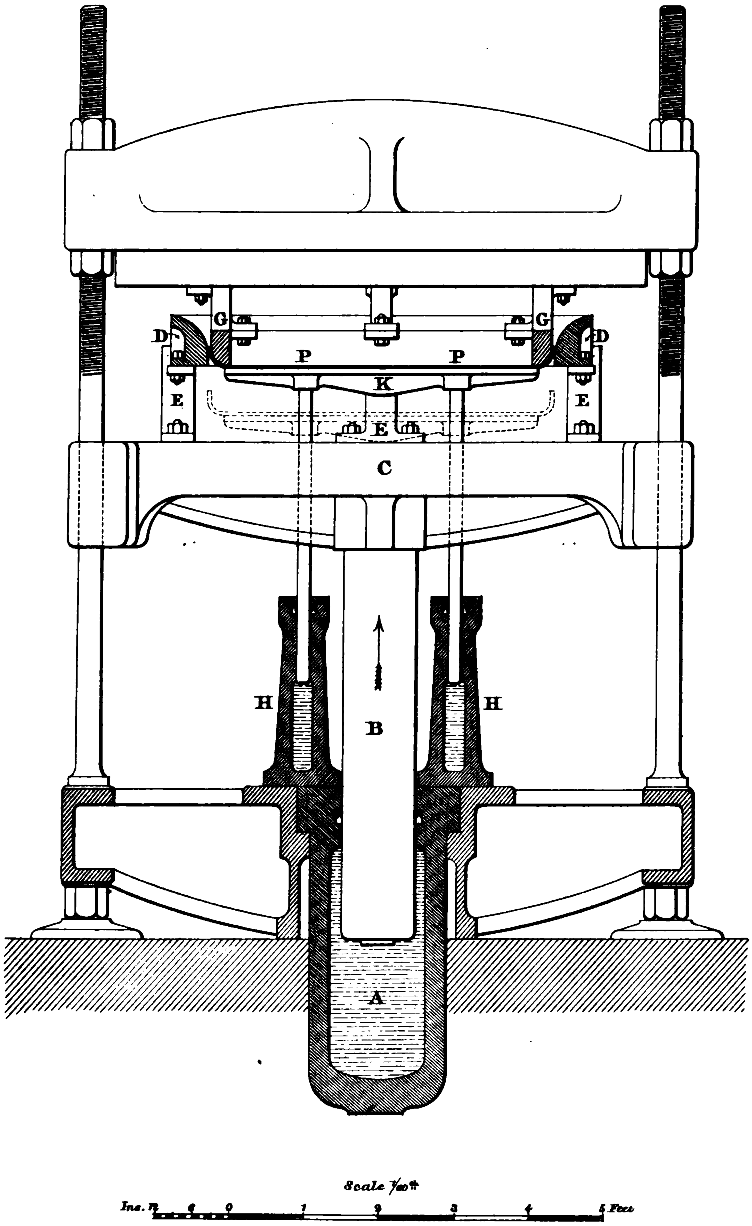
The gas occluded in steel ingots is chiefly hydrogen, which is supposed to be dispersed through the mass, owing to the decomposition of the moisture in the air that is blown in. The removal of the occluded gas has been the object of several processes; in one the molten mass is subject to a pressure of steam whilst it is setting, the power of rapid diffusion of the gas through red hot steel enabling it to be thus removed.

The employment of hydraulic pressure for the manufacture of steel guns was referred to by Major Mackinlay, R.A., in a paper which he read at the Royal United Service Institution in 1885. Detailed information on this subject is not at present readily obtainable, owing to a reluctance to make public the practical points which have to be observed to secure success. The construction of a steel gun by the aid of hydraulic presses engaged Sir Joseph Whitworth's attention, and he applied to this purpose the principle which he had successfully used to manufacture hollow propeller shafts. In this case the solid cylindrical ingot from which the shaft is to be made is first bored and converted into a hollow cylinder. It is then heated, and a hollow steel mandril of smaller diameter than the interior is placed inside it, and the action of hydraulic pressure is brought to bear upon the external longitudinal surface of the cylinder. The press squeezes the metal against the mandril within (which is kept cool by water flowing through it), the cylinder being turned over during the operation, so that it is evenly pressed throughout. The effect of this pressing is to bring the internal diameter of the cylinder to that of the mandril, and at the same time the length of the cylinder is increased. By reheating the cylinder, and repeating the process of pressing with smaller mandrils, the final proportions of the propeller shaft are ob-

tained. A similar process is employed in making steel guns and presses. The ingot is cut into thick rings which are squeezed in presses round mandrils, as already described.

In applying hydraulic power to flanging and bending plates, it was found, after long trials, that when solid dies were used the cost of the dies and moulds was excessive, but that this difficulty would be got over by means of hollow dies. Mr. Gustave Piedbœuf, of Jupille, near Liège, explained to the Institution of Mechanical Engineers the construction of a hydraulic flanging machine which is shown in Plate 20. This machine has a hollow ring die which is very well adapted to its purpose. The cylinder A is connected to an accumulator loaded to about 1500 lbs. per square inch, and the water, acting on the ram B, raises the moving table C, carrying with it the matrix or annular die D, which is supported on small columns E. Some of these are not bolted to the table, but are only slipped in, and can thus be easily taken out when the plate P has been flanged, and requires to be removed. The block G is attached to the top frame, and corresponds to the matrix; and although in this example a round plate is shown, the block and matrix can be made to any required shape. The small auxiliary cylinders H carry on their rams a table K, suitable openings being made in the main table C to allow of the rams H moving freely up and down independently. The full black line in the drawing shows a plate P at the moment before completion. The dotted lines show it just completed, and detached from the block G, and ready to be removed by taking out one of the columns E. To flange a plate after heating it, the main table C, and small table K, are let down to their bottom position, and the plate being placed on the table K, pressure is admitted to the cylinders H, causing this table to rise up and hold the plate against the block G, which prevents all buckling and risk of unequal flanging. Pressure is then admitted to the main cylinder A, causing the main table C (carrying the matrix D) to rise; the matrix catches the edge of the plate P, and forces it over the block G, as shown. The water is then let out of the

HYDRAULIC FLANGING MACHINE.



Thos. Kell & Son, Lith.

cylinders, and the plate is done, the whole operation occupying only about half a minute. In using this machine it is necessary to heat the whole plate, and to effect the flanging at one stroke, which involves the use of furnaces and machinery, as well as of blocks and dies large enough to suit different sizes and forms of plates.

Messrs. Tweddell, Platt, Fielding, and Boyd have introduced

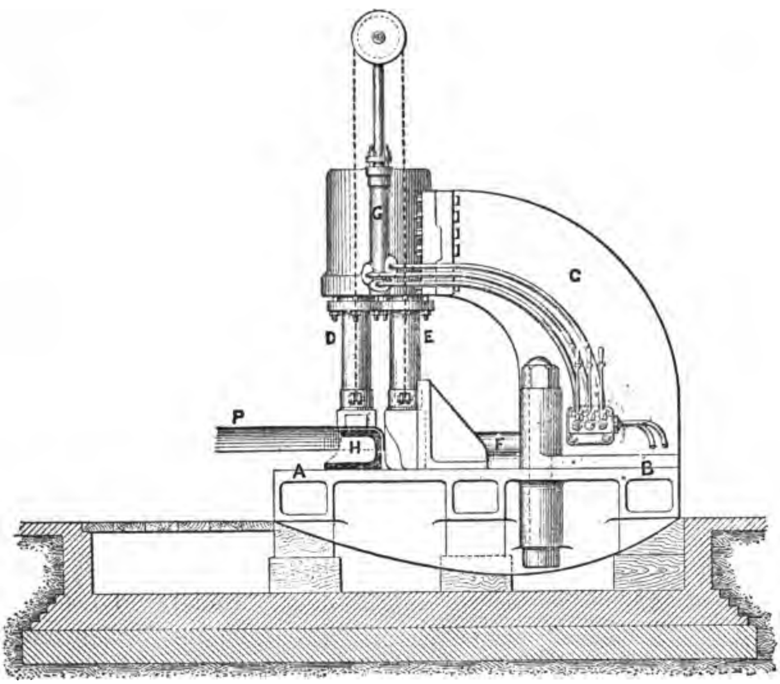


FIG. 34.

a machine, which was described in a paper read at the Institution of Civil Engineers, and is shown by figs. 34 and 35. By means of this machine the flanging can be effected by several successive operations. The tools employed are simple, and of comparatively small dimensions, and the power exerted (which has to perform only a fraction of the work at a time) is much less than when the whole has to be done at once. In fig. 34 AB is a bed plate, at one end of which is fixed a frame C, carrying three

hydraulic cylinders having plungers D, E, and F, on the ends of which are fixed suitable tools. The two plungers D and E, working vertically, may be drawn up by means of chains, worked by the plunger of a drawback cylinder. Should the plate to be flanged be circular, a turntable is placed in a suitable position, and the anvil block H is also adjusted to the required radius of the flange. The plungers D, E, and F being withdrawn, the plate, which has been heated at the edge, is placed with its edge projecting beyond the anvil H. The plunger D is then lowered upon the plate, so as to press it firmly on the anvil, and thereupon the plunger E (which carries a tool suitably sloped on its face) descends, and bends part of the edge of the plate P over the anvil H. The plunger E being now raised, the horizontal plunger F is advanced, so as to press the bent part of the plate against the face of the anvil H. The plungers being again withdrawn, the plate P is turned partly round, so as to present a fresh portion of its edge, which is similarly operated on. Various modifications have been introduced into the machine to suit different classes of work. It can be used for flanging plates in the ordinary way, the two plungers acting together on the moving block K, as shown in fig. 35.

M. Berrier-Fontaine has given the following results, which he obtained from a plate-flanging machine at the Toulon Dockyard. Effective diameter of ram, 19·39 inches. Effective area of ram, 186 square inches. Maximum stroke of ram taken by machine, 2563·05 cubic inches. Expended from accumulator, 2849·87 cubic inches. Effective pressure transmitted by ram, 118·10 tons. Maximum work per stroke of ram done by machine, 303,786 foot pounds. Expended by prime mover, 445,679 foot pounds. Maximum per day of 10 hours, 50 strokes of ram, 82·39 cubic feet of water expended, and 22,783,950 foot pounds of work expended.

In November 1882 Messrs. Easton & Anderson made some experiments on flanging cold steel plates by hydraulic pressure. A pair of moulds were made and fitted to a hydraulic press capable of exerting a pressure of about 250 tons. They were

so shaped that at one operation they made a flange both on the outer and inner edges of an annular steel plate, and thus produced a double-flanged annulus. The upper mould should be formed concave, and the lower convex (to the extent of $\frac{1}{16}$th of an inch), in order to flatten the face of the plate. The plates experimented on were Landore Siemens, of S and SS

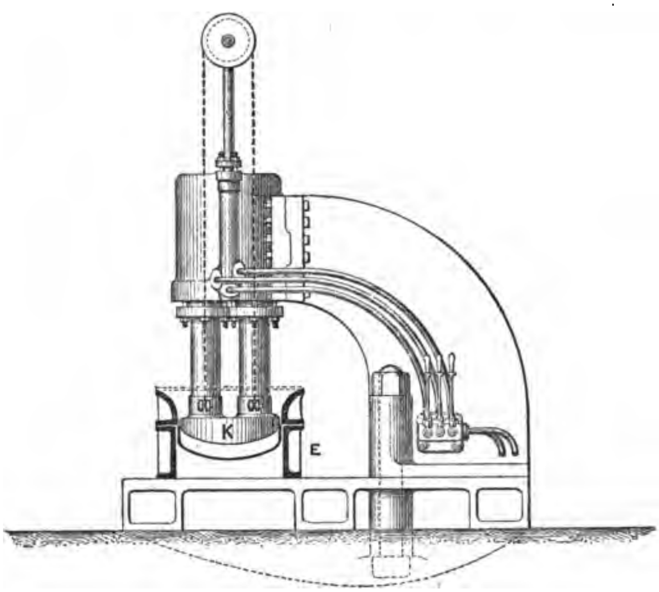


FIG. 35.

quality, $\frac{3}{8}$ths of an inch thick. Some were annealed and others were not.

Messrs. Hugh Smith & Co. of Glasgow have arranged a hydraulic machine for flanging, rivetting, &c., by which pressure from a cylinder acts with an increasing leverage towards the end of the stroke, when the maximum pressure is required.

In a blooming mill at Ebbw Vale, a pair of hydraulic rams under constant pressure are employed to counterbalance the weight of the top roll, by being placed vertically beneath the bed plate of the roll frames, one under each frame. The rams are kept under a pressure of 450 lbs. per square inch, and the diameter being $10\frac{3}{8}$ inches, a constant lifting force of 34 tons

is exerted on the top roll, so raising it, and facilitating the entering of the ingot at each passage through the rolls. A tightening down screw, connected with a pinion and toothed quadrant, and actuated by a horizontal hydraulic press, is employed to lower the top roll. One revolution of the pinion suffices for the whole vertical range of the roll. One movement of the handle which controls the admission of water to the press for lowering the top roll, serves to slacken the tightening down screws, the top roll being raised automatically by the pressure on the rams supporting it.

At the Elswick Works a forging press is constructed with a cylinder 35 inches in diameter, to work at 3 tons per square inch, so that a pressure of about 3000 tons is exerted. The stroke of the ram is 9 feet, which provides for punching the centres out of ingots. The work of punching an ingot 6 feet long is done in two operations, and an ingot 4 feet long in one operation. This press is served by four gas-reheating furnaces, and two overhead hydraulic cranes capable of handling 60 tons. The total cost was about £25,000.

Messrs. Vickers & Co. of Sheffield are erecting a forging press, with cylinders 43 inches in diameter and about 5 feet stroke.

Messrs. Davy Brothers of Sheffield are constructing a 4000-ton hydraulic forging press for Messrs. Charles Cammell & Co. of Sheffield. In this machine two presses are employed instead of one, with the resulting advantage of lessening the weights and strains on the parts. The moving cross-head carrying the tool is guided by slide-blocks bored to fit the wrought steel columns which support the entablative on which the presses rest. By this arrangement the cross-head is guided independently of the pressing rams, and a forging can thus be placed considerably out of the mid-position between the rams, whilst the width across the entablative can be reduced to the smallest limit. The sling chains holding the work are therefore able to be brought closer than usual. A low-pressure water service of 60 lbs. to the square inch is connected to the presses, and fills

them as they descend upon the forging, when the high-pressure pumps start, and automatically cut off the low-pressure. The relative areas of the lifting and pressing rams are as 16 to 1. The pumps at each revolution depress the cross-head half an inch, whilst the lift is 8 inches. The rapidity of the pressing and return stroke can be made considerable by running the pumps quicker. The tool is raised at each operation about 6 inches above the forging, to allow it to be moved. This clearance is taken up at the rate of 2 feet per second, by connecting the main ram cases with the low-pressure service. There are two main-pressure rams 36 inches in diameter, and two lifting rams 9 inches in diameter, the ram cases being made of wrought steel. The stroke of all the rams is 7 feet. The maximum hydraulic pressure is 4500 lbs. to the square inch, and this is obtained from a set of three single-acting ram pumps 6 inches in diameter and 12-inch stroke, which are driven from the crank-shaft of a steam-engine.

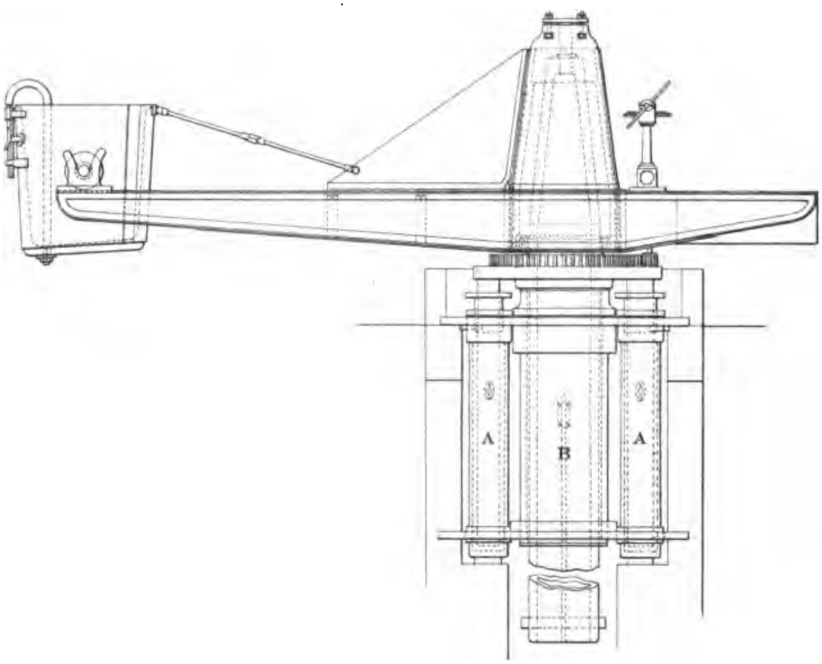
In the Bessemer and Siemens processes for the manufacture of steel, the ingot cranes have to be quickly and easily controlled, with a freedom from gearing, and for this purpose hydraulic power has been adopted as the most convenient. The cranes which are required for the purpose of manufacturing steel have to deal with a different load, and with a different condition of things, to that which formerly existed, as the modern converters produce at each "blow" as much as 12 to 15 tons, compared with a quarter of that weight which was obtained previously. In steel-making, the centre crane is relied on to receive, and distribute, the molten steel in the process, and it has to move a ladle, which must be capable of motion in several directions, in order to accommodate itself to the tipping motion of the converter, which involves a vertical and horizontal motion of a few feet. It must also command a considerable horizontal radius, so as to deliver into the various ingot moulds. The vertical motion is capable of being accomplished by employing a hydraulic ram working vertically in a cylinder, and having a jib or beam resting on the top of, and

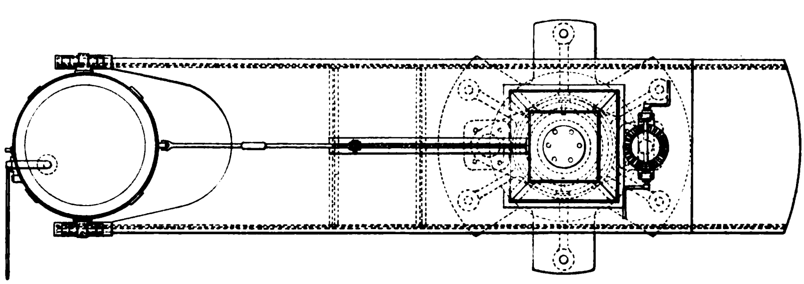
rising and falling with, the ram. The beam or jib has the ladle at one end, and a counterweight at the other. The horizontal motions are obtained either by hydraulic power, or by hand-gearing.

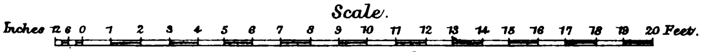
Sir Henry Bessemer originally gave considerable attention to the arrangement of the machinery for the pits of steelworks for his process, and, in principle, this remains the same now. Attention has been of late years directed to effecting economies in the production of hydraulic power, by employing compound engines, and by saving water during the working of the crane, when the loads to be raised and lowered vary. Also by minimising the effect of the great leverage of the weighted ladle on the beam. In the original forms of crane one centre ram was made large enough to lift both the ingot and its own weight, together with that of the top and jib of the crane, and to act also as a guide. A more modern arrangement is shown by Plate 21, which is a "Hydraulic Centre Crane" by Messrs. Tannett, Walker, & Co., of Leeds, capable of lifting 12 tons. The radius is 17 feet, and the stroke or vertical lift is 7 feet. There are two rams AA (9 inches in diameter), in communication with the accumulator, nearly balancing the dead load of the rams, jib, ladle, &c. Sufficient margin is allowed to admit of the crane descending and forcing back into the accumulator the water that is displaced by this ram. A third ram B (12½ inches in diameter) is under the control of the workman by means of a valve, which can be placed either close to the crane or at a distance from it. In this case the turning is done by hand, but it is often performed by a hydraulic engine. The two side rams effect a saving of water-power, and enable the crane to be made of any desired strength, without involving the loss of power that occurred in the old form of crane.

Mr. Thomas Wrightson has designed a good form of balance crane to meet the difficulty of dealing with the strains in a heavy 15-ton casting crane. This is shown by fig. 36. The crane post revolves on a pivot, and carries the cylinder with it, in its horizontal rotation, by means of a key. Frames for the

12 TONS HYDRAULIC CENTRE CRANE.







Thoˢ Kell & Son, Lith.

sheaves are fixed on the ram, and revolve with it. The top support for the ram is attached to the roof, the maximum horizontal strain not exceeding $4\frac{1}{2}$ tons. The lifting cylinder has a 21-inch gland at the bottom, and a 12-inch gland at the top. This cylinder works up and down upon the post, the top gland of the cylinder working in the smaller diameter, and the bottom gland working in the larger diameter, of the post. Thus

15-TON CASTING CRANE.

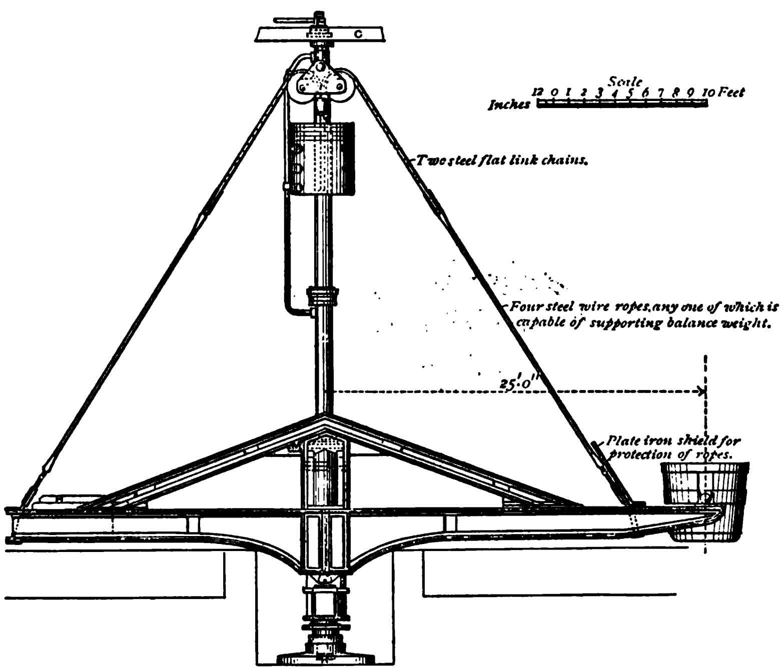


FIG. 36.

when water is admitted into the cylinder (through a hole in the post) the cylinder itself rises with a lifting power equal to the difference of the two areas of the post, multiplied by the effective pressure of the water. Further, by flattening one side of the post at the larger diameter, and adapting the lower gland-box to this form, a sliding-key arrangement is produced, so that for horizontal rotation, the cylinder and post move round

together. The machinery and ladle are carried by a cradle with a pivot fixed at the bottom of the cylinder, on which the cradle has a slight rocking movement. The ladle when full contains 12 tons of steel. A weight is adjusted so as to balance the cradle when the ladle contains half its charge, or 6 tons of steel. A revolving port on the top of the crane post conveys water to and from the cylinder. A stop is provided to prevent the cradle from drooping when the preponderance of weight is on the ladle end, but if the bottom of the ladle be lowered upon anything unyielding, the frame simply hinges upwards on its pivot. The pipe conveying water to and from the cylinder is arranged so that the lower end terminates in a hole which is bored through the centre of the ram, the upper end also entering the post to convey water through the top bearing.

It will be seen that provision has to be made to balance the half weight of the steel, which is not balanced by the counterweight. To accomplish this, chains are led from each end of the girders forming the platform, over sheaves fixed on a strong frame at the top, and forming part of the crane post immediately under the top socket, so that the sheave frame can rotate horizontally with the crane post and cylinder. The two sets of chains, after passing over their respective sheaves, descend to a heavy balance weight of annular form, surrounding the upper portion of the crane post, which acts as its guide. The point of connection of both sets of chains is the same, and is in a plane passing through the centre of gravity of the weight, so that it may hang indifferently on either one or the other set of chains. The chains are a succession of flat steel links connected by pins.

If the ladle is half full, then the fixed counterweight balances this amount of steel, and the annular balance distributes its weight equally between the two sets of chains, neutralising so much of the dead weight of the platform, and so saving water pressure in the cylinder. If the ladle is full, the half weight tends to bring down the ladle end of the platform, but is prevented (owing to its rigidity), without the opposite end being raised to an equal extent. The depression of the ladle end,

therefore, tightens its chains, whilst the elevation of the opposite end slackens its chains. By this means the whole weight of the annular balance comes on the tight chains of the ladle end, and thus any preponderant weight in the ladle is balanced automatically.

As the steel is run into the ingot moulds the preponderance becomes less, until when more than half is run out, the preponderance is transferred to the opposite end of the platform. As this takes place, the opposite chains are tightened by the action of the fixed balance weight, until, by the time the whole of the steel has run out of the ladle, the entire weight of the annular balance is hanging on the set of chains opposite to the ladle, and in fact balances the whole effect of the fixed weight on the platform. This transmission of the forces is entirely automatic; the annular balance, by means of this special mechanical arrangement, divides its weight between the two ends of the platform, in the exact proportions required to maintain equilibrium, and this without affecting any of the other motions of the crane, which may be going on at the same time.

Mr. B. Baker (the President of the Mechanical Science Section of the British Association in 1885) acknowledged the important part that hydraulic appliances had played in the construction of the Forth Bridge in the following words: "More than 42,000 tons of steel plates and bars have to be bent, planed, drilled, and rivetted together before or after erection, and hydraulic appliances are used throughout. The plates are handled in the shops by numerous little hydraulic cranes of special design, without any complication of multiplying sheaves, the whole arm being raised with the load by a 4-inch direct-acting ram of 6-feet stroke. A total length of no less than 60 miles of steel plates, ranging in thickness from 1¼ inch to ⅜ inch have to be bent to radii of from 6 feet to 9 inches, which is done in heavy cast iron dies squeezed together by four rams of 24 inches in diameter and the same stroke. With the ordinary working-pressure of 1000 lbs. per square inch, the power of the press is thus about 1750 tons. Some

3000 pieces, shaped like the lid of a box, 15 inches by 12 inches wide, with a 3-inch deep rim all round, were required to be made of ½-inch steel plate, and this was easily effected in two heats by a couple of strokes of a 14-inch ram." He also described that, in erecting the great 1700 feet spans of that bridge, the massive girders were put together at a low level, and were hoisted as high as the top of St. Paul's Cathedral, by hydraulic power. Continuous girders, nearly one-third of a mile in length, were similarly raised, together with the necessary sheds, cranes, appliances, and workmen, the whole weight on the platforms being in some instances more than 1000 tons.

In the excavation of the foundations of the Forth Bridge hydraulic appliances of a novel kind were used. The huge wrought iron caissons (70 feet in diameter and 70 feet high) for the foundations had to be sunk through tenacious boulder clay, which was excavated by hydraulic spades. Hydraulic rams worked in the hollow handles, which were thrust against the roof, and by turning a tap the spade was forced into the clay, with a pressure of three tons. These hydraulic spades were employed in an electrically lighted diving-bell 70 feet in diameter, 7 feet high, and 90 feet below the sea.

CRANES.

In all the designs for hydraulic cranes, the principle employed is that of using the direct thrust of a ram or piston through a short stroke, and multiplying the stroke by carrying the lifting chain over a series of sheaves. In general, the cylinders and machinery are placed horizontally in a chamber underground. In some cases the lifting cylinder is placed vertically, and is made to form part of the pillar of the crane, as is shown by Plate 22, fig. 1, which represents a Goods Station Crane for a lofty goods shed, the pillar being carried by top and bottom bearings. The lifting cylinder is placed in the pillar of the crane, to

Figs: 1.

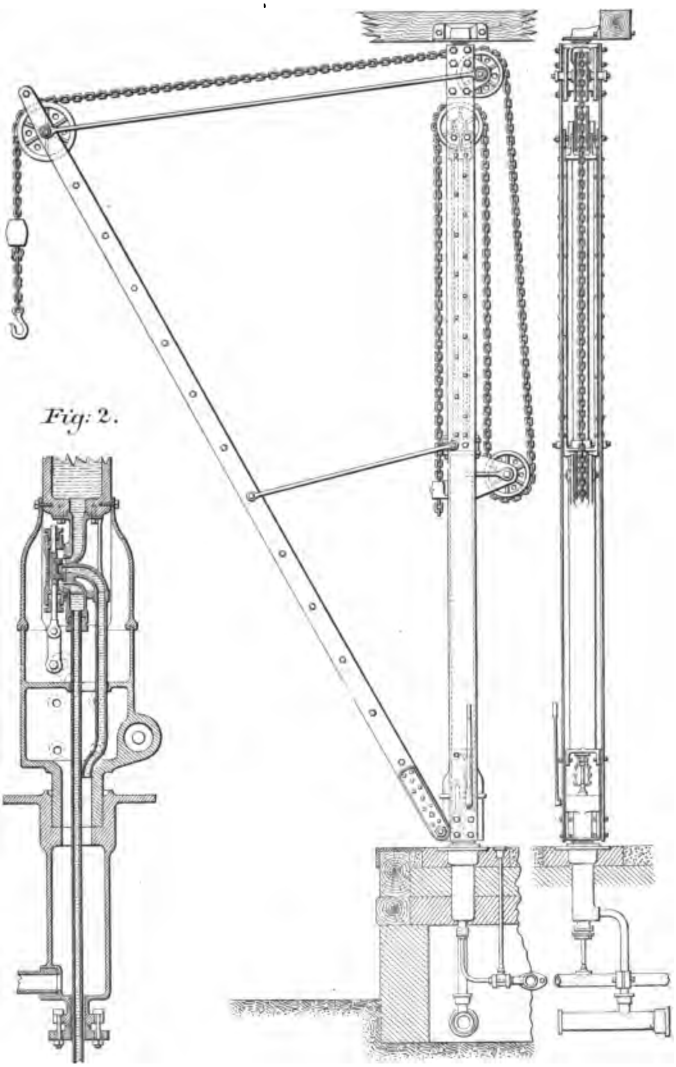
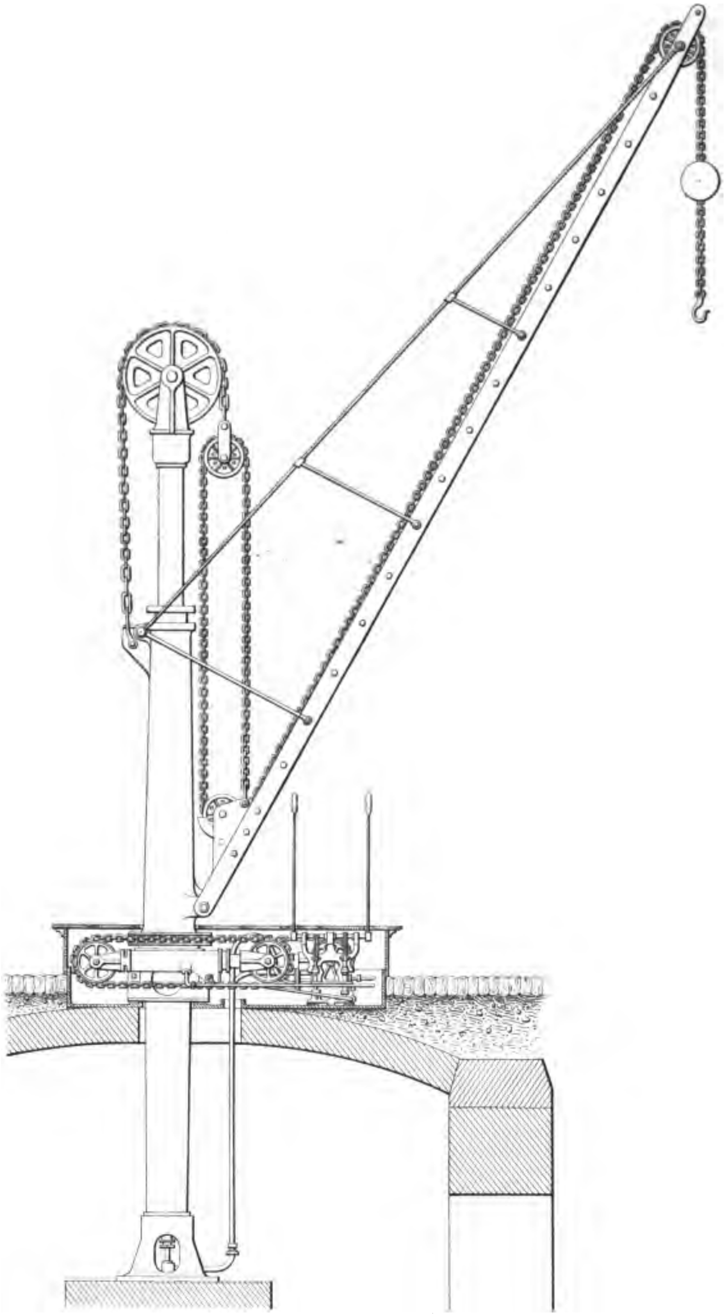


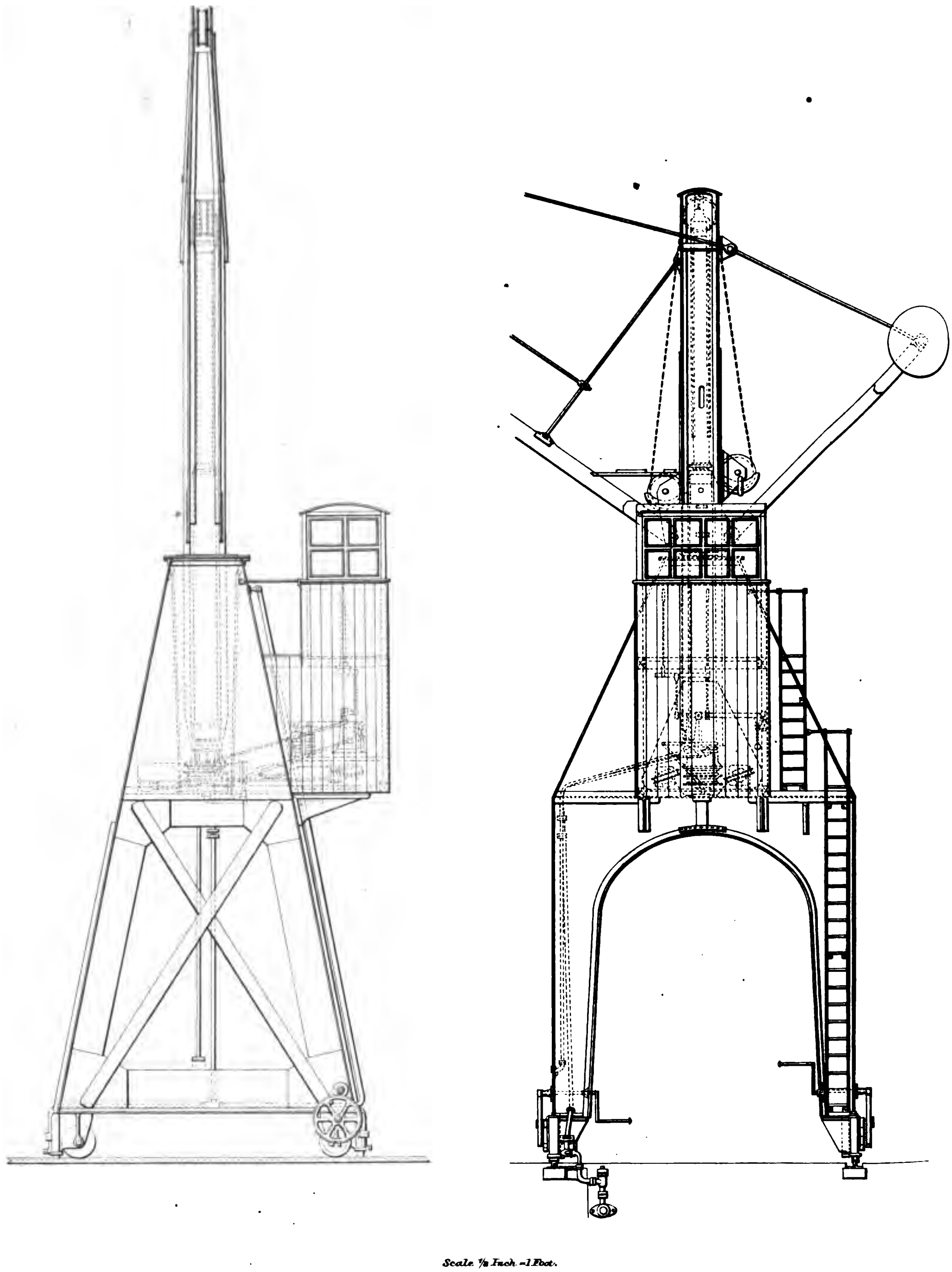
Fig: 2.

Thos. Kell & Son, Lith.



Thoˢ Kell & Son, Lith.

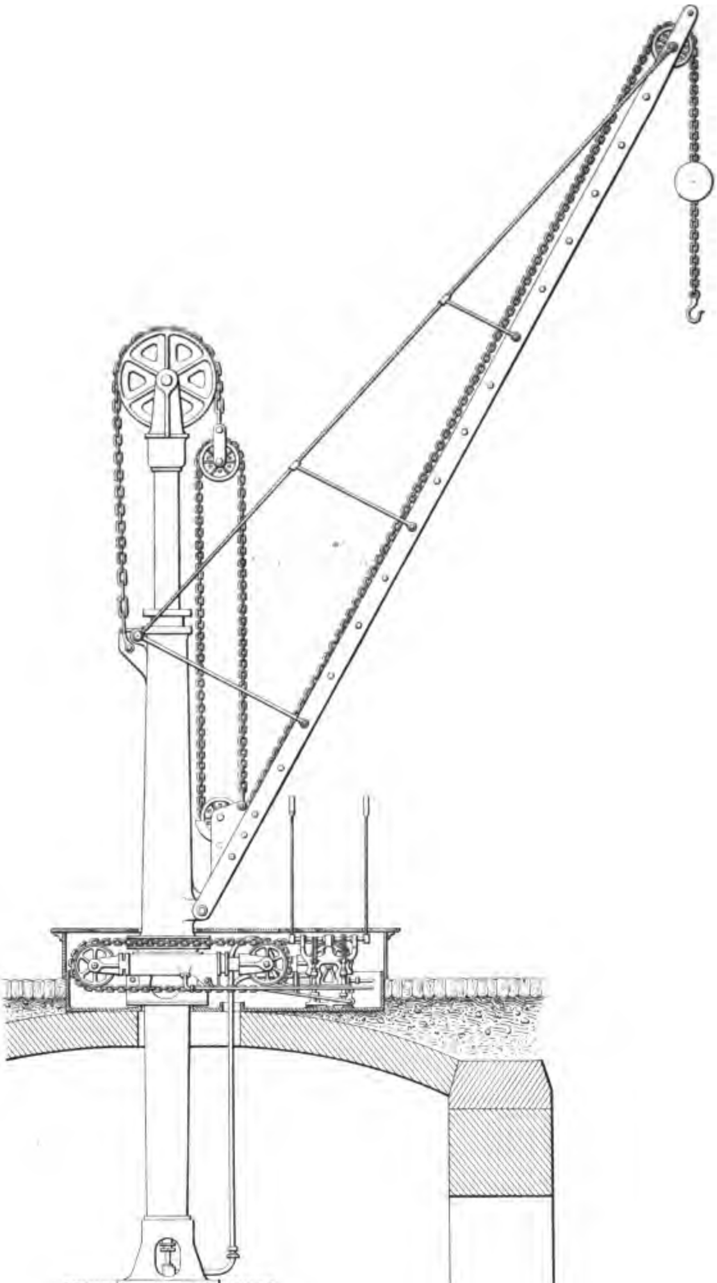
MOVEABLE HYDRAULIC CRANE.



Scale ½ Inch = 1 Foot.

Inches 12 6 0 1 2 3 4 5 10 15 20 Feet

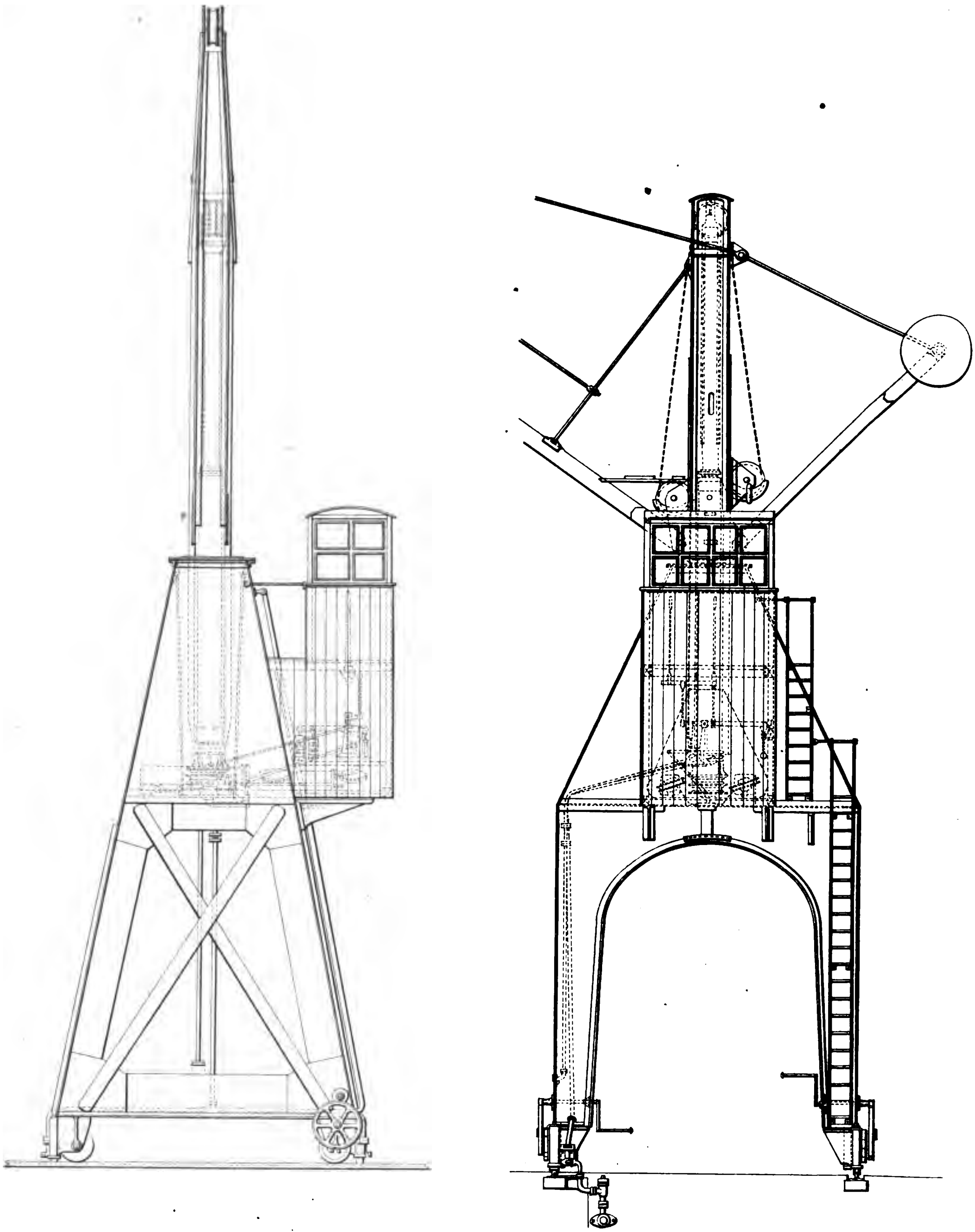
Thoˢ Kell & Son, Lith.



Thos. Kell & Son, Lith.

PLATE. 24.

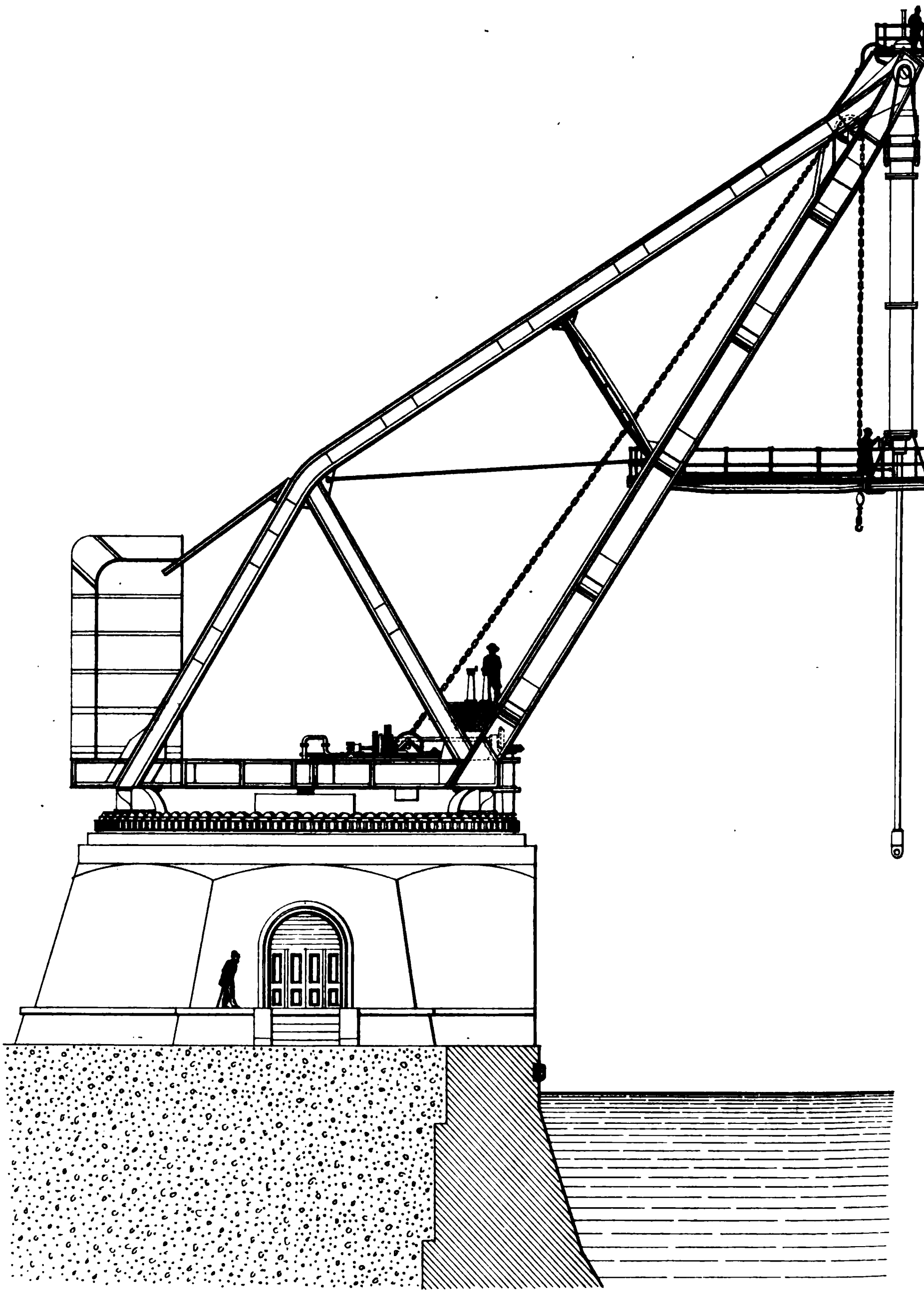
MOVEABLE HYDRAULIC CRANE.



Scale 1/8 Inch = 1 Foot.

Inches 12 6 0 1 2 3 4 5 10 15 20. Feet

Thos Kell & Son, Lith.



Thoˢ Kell & Son, Lith.

which pillar the working valve is fixed, the water entering and escaping through the pivot as shown by fig. 2.

A form of station crane is shown by Plate 23, which represents one of the cranes erected at the new goods station of the North-Eastern Railway at Newcastle-on-Tyne. These types of cranes were explained by Sir William Armstrong in a paper read at the Institution of Civil Engineers.

Plate 24 shows an Elswick "Moveable Hydraulic Crane."

When the load to be raised becomes very great (100 tons or so), it is better to substitute some other arrangement for that of chains. In the case of a large crane which Sir William Armstrong, Mitchell, & Co. erected at the Royal Italian Arsenal at Spezzia, the lift is performed by the direct action of a piston contained in an inverted cylinder suspended in gimbals from the head of the jib, as shown by Plate 25. This crane is capable of lifting 160 tons through a range of 40 feet. It is carried upon a ring of live rollers supported by a pedestal of masonry, and the slewing is effected by a hydraulic engine applied to a pinion gearing into a circular rack. The jib projects 65 feet from the centre of rotation, and its height above the quay level is 105 feet. If the crane is used to lift much lighter loads than the maximum, a chain is employed, which is raised and lowered from a cupped drum, worked by the slewing engine.

HYDRAULIC POWER APPLIED TO BRIDGES.

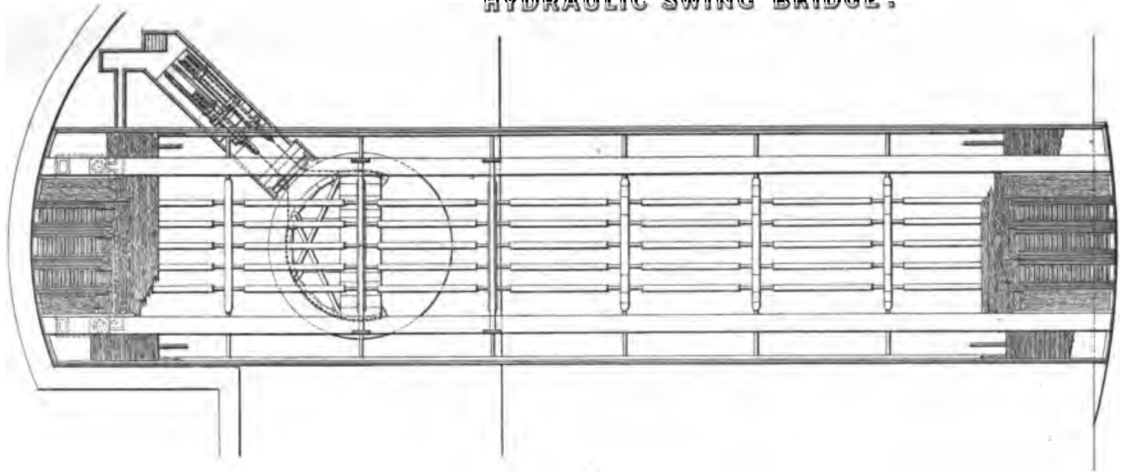
The first application of hydraulic power to bridges was in 1852, when the Forest of Dean Railway Company constructed a hydraulic swing bridge over the river Severn. A double leaf swing bridge of timber was constructed and worked by hydraulic power about the same time at the Birkenhead Docks. Each leaf of this bridge had a central hydraulic press of sufficient power to lift the leaf, and acting at the same time on the pivot upon which the bridge revolved. The tail end of the bridge

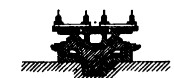
was fitted with two rollers, which were brought in contact against an upper rail, so that, as the tail end was lighter than the nose end, when the bridge was lifted from its bearings, the rollers at the tail end were brought against the rail, and the pressure, being continued on the centre press, lifted the bridge clear of the masonry. The swinging of the bridge was effected by means of a hand rack and pinion, but as the bridge was carried on a water pivot it was easily worked. The bridge rested on masonry supports when not at work. The first operation of lifting the bridge off its bearings was performed in a few seconds, by turning the water from the accumulator into the centre press.

In some examples of the application of hydraulic power to swing bridges a single press forms the water-bed. In others the ram is not attached to the bridge, but the end of it is made of a cup-shape, in which revolves the pivot that is attached to the bridge. In some swing bridges the tail end is made light. In others the tail end of the bridge is made heavy, so that when the bridge is lifted, the nose end is raised, the tail end resting upon a rail below the roller path. Occasionally it has been found advisable (as a stand by) to have a means of working the bridge by hand power. This is done by using the centre press as a solid pivot, the ram resting upon the bottom of the cylinder, and the bridge revolving in a cup formed in the upper part of the ram. By means of presses applied to the tail end of the bridge, and worked by hand-pumps, it can be lifted or lowered to clear the bridge from its resting-blocks, and to leave it free to revolve.

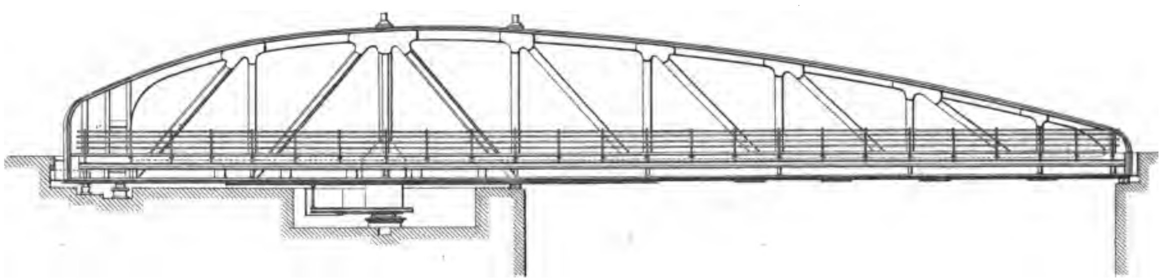
Plate 26 shows a "Hydraulic Swing Bridge" constructed by Sir William Armstrong, Mitchell & Co. This bridge crosses a clear opening of 100 feet, and is adapted for both road and roadway traffic. The clear width of roadway between the kerbs is 23 feet. The footways are on the outside of the main girders, and have each a clear width of 4 feet 10 inches. The bridge is designed for a rolling load of $1\frac{1}{4}$ tons per foot run on each line of railway, and for a concentrated load of 60 tons on

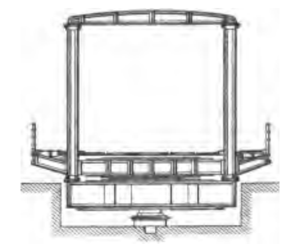
HYDRAULIC SWING BRIDGE.

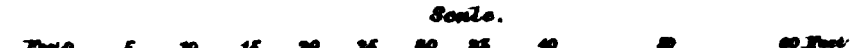




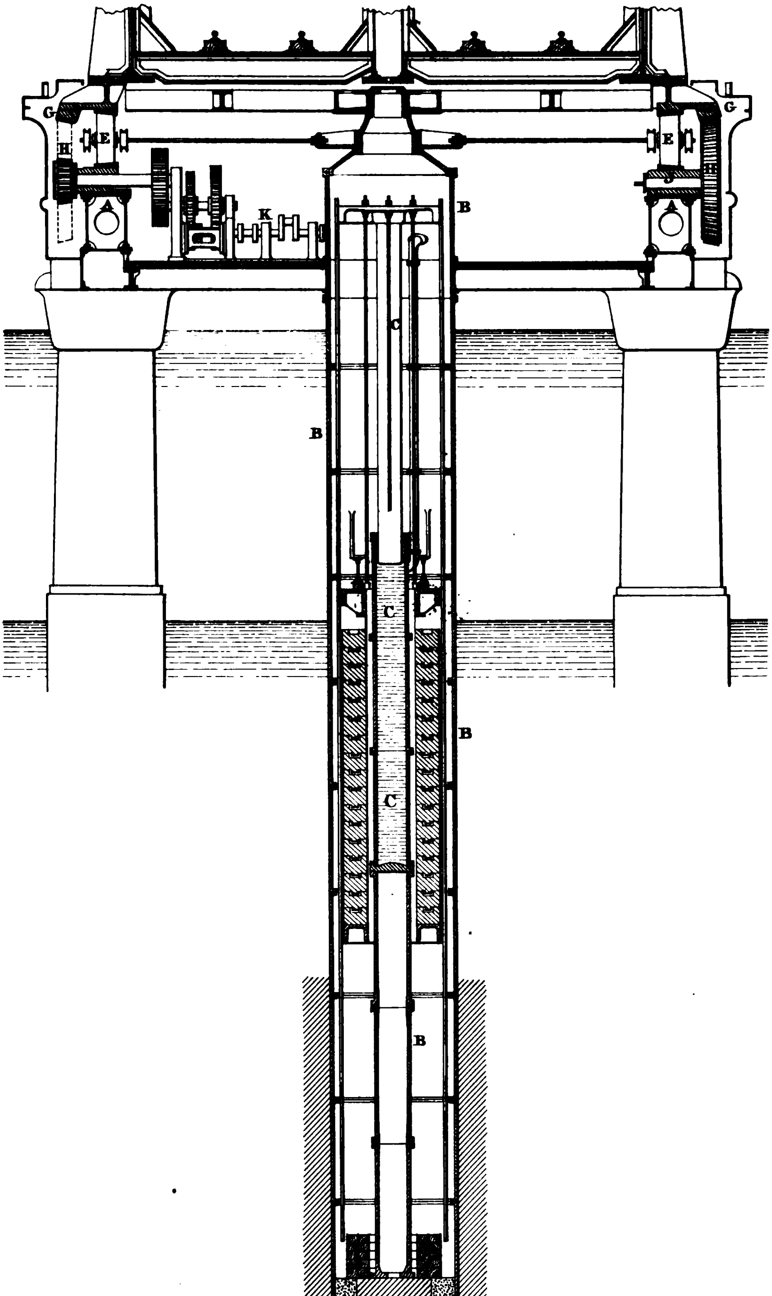
SECTION THROUGH CENTRE PRESS.







Thos Kell & Son, Lith.

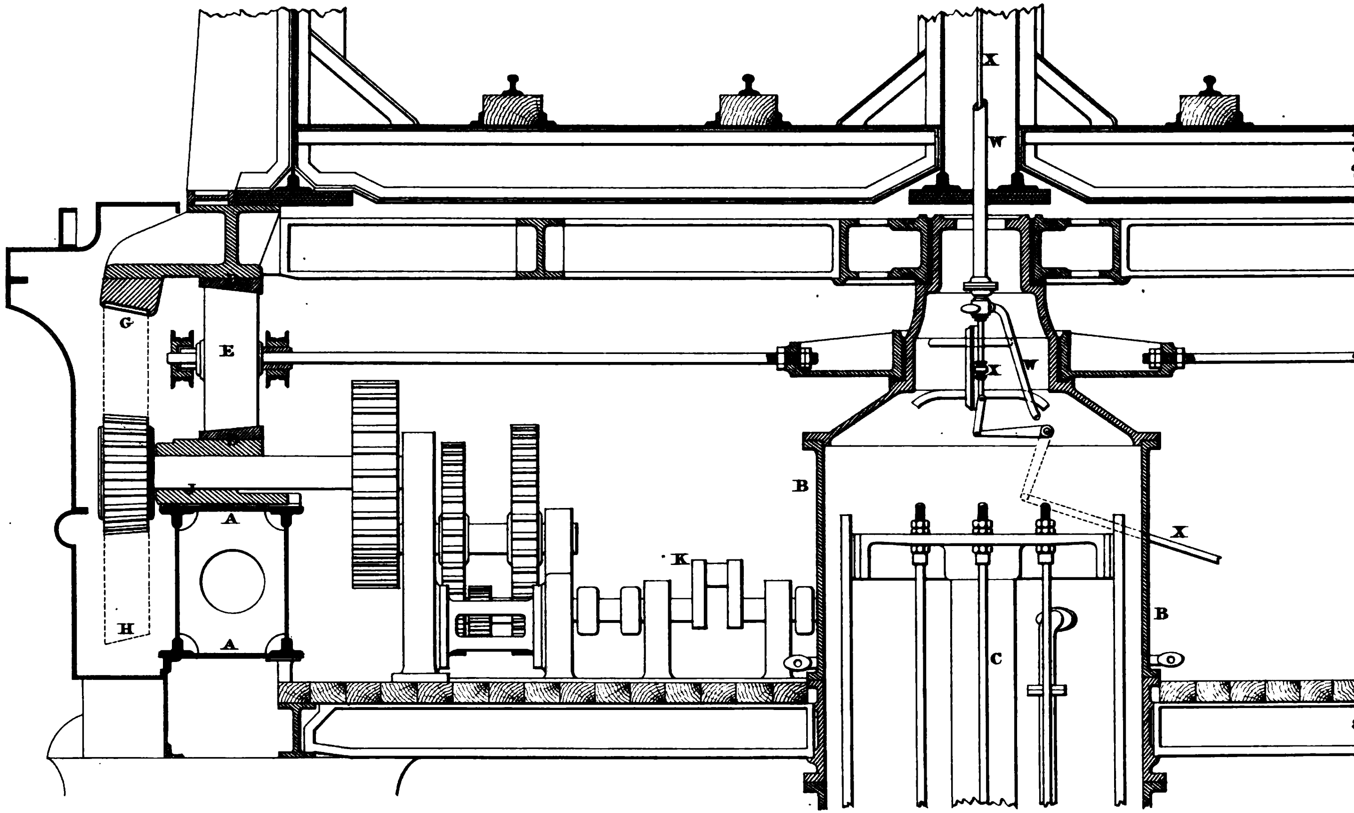


SECTION OF CENTRE PIER AND ENGINE ROOM.

Scale 1/120th
Feet 10 5 0 10 20 Feet

Thos Kell & Son, Lith.

OUSE SWING BRIDGE.



ENLARGED SECTION OF ENGINE ROOM.

Scale 1/48th

Ins. 12 6 0 1 2 3 4 5 6 7 8 9 10 Feet

Thos Kell & Son, Lith.

four wheels. The main girders are of the triangular or braced construction, the cross-girders for carrying the roadway being fixed to the main girders at the foot of each system of triangulation, with longitudinal girders between the cross-girders under each line of rails. The bridge is lifted from its bearings (preparatory to being turned round) by an hydraulic press acting on a wrought iron bearing-girder fixed to the underside of the main girders. The turning motion is effected by two hydraulic cylinders acting, by means of chains, on a turning drum fixed to the under side of the bridge. A hand-pump is provided for use, in case the pressure from the main is not available. Should the centre press become disabled, provision is made for tilting the bridge from its bearings, by lowering the rear end wedge-resting block, hand-presses being provided for this purpose.

Another form of swing-bridge, to which hydraulic power is applied, is that which rests upon a circle of live rollers, on a permanent roller path. The bridge is made to revolve either by means of one or more rotary hydraulic engines, placed on the centre pier within the circle of rollers, or by means of reciprocating acting rams and chains attached to the drum in the centre. The ends of each main girder are blocked by hydraulic presses after the bridge is closed.

The railway bridge over the river Ouse at Goole, which is one of the most important examples of this kind, was described by Sir William Armstrong, when President of the Institution of Mechanical Engineers in 1869, and is shown by Plates 27 and 28. This bridge carries a double line of railway across the river Ouse, by means of three wrought iron plate girders, which, for the swinging portion of the bridge, are 250 feet long and 16 feet 6 inches deep in the centre. Plate 27 is a vertical transverse section at the centre pier, showing the engine-room and accumulator situated within it. Plate 28 is an enlarged section of one half of the engine-room. The centre girder of the three is strengthened, and rests on an annular box girder AA, 32 feet in diameter, which forms the cap of the

centre pier. This cap rests on the top of six cast iron columns, 7 feet in diameter, which are arranged in a circle and form the centre pier. A centre column BB, also 7 feet in diameter, contains the accumulator, and is attached to the others by cast iron stays which support a floor. On this the steam-engine, boilers, &c., for producing the hydraulic power are fixed.

The weight of the swing-bridge is 670 tons, and it rests entirely on a circle of 26 conical live rollers, EE. These are 3 feet in diameter, with 14 inches width of tread, and run between the two circular roller-paths, DD, which are 32 feet in diameter and 15 inches wide.

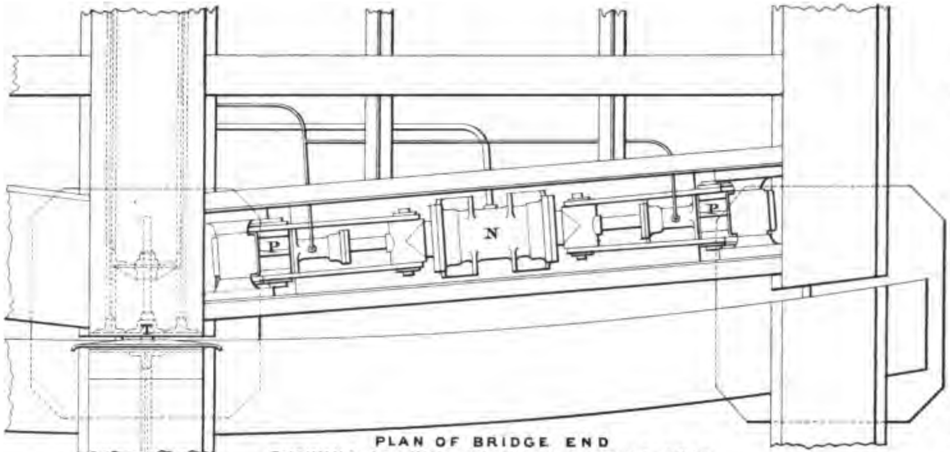
The turning motion is communicated to the bridge by a bevil wheel H, which gears into a cast iron circular rack, G, bolted to the outer circumference of the upper roller path. A steel pin J, supported in the lower roller path, carries the bevil wheel. This is driven by a pinion connected by intermediate gearing with a three-cylinder hydraulic engine (in duplicate) placed at KK, which exerts a force of about 10 tons at the radius of the roller path. The engines work at 40 revolutions per minute, with a water-pressure of 700 lbs. per square inch. The power is obtained from a pair of 12 HP steam-engines fixed (as before stated) in the engine-room formed beneath the centre of the bridge. Water is delivered into the accumulator C, which has a ram 16½ inches in diameter and 17-feet stroke, and is loaded with a weight of 67 tons.

To secure a solid roadway, and a perfect continuity of the line of rails, an arrangement of gearing, shown by Plate 29, figs. 1, 2, 3, and 4 is used. By this each extremity of the bridge is slightly lifted by a horizontal hydraulic press N, acting on the levers PP, forming a "toggle joint." The press has two rams acting in opposite directions upon two toggle joint-levers, connected by a bar Q, which moves in a vertical guide to insure a perfectly parallel action of the two points. By this means the end of the bridge is made truly parallel when the resting-blocks RR, under each girder, are put into position. To do this, three separate hydraulic cylinders, SS, are employed, as shown by figs.

OUSE SWING BRIDGE,

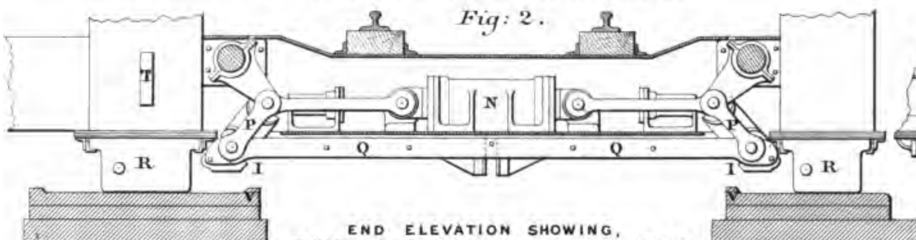
LOCKING GEAR &c. AT ENDS OF BRIDGE.

Fig: 1.



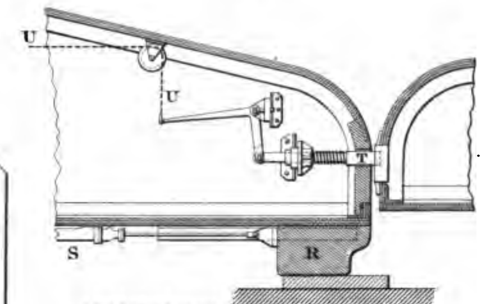
PLAN OF BRIDGE END
SHOWING LOCKING BOLT AND LIFTING GEAR

Fig: 2.



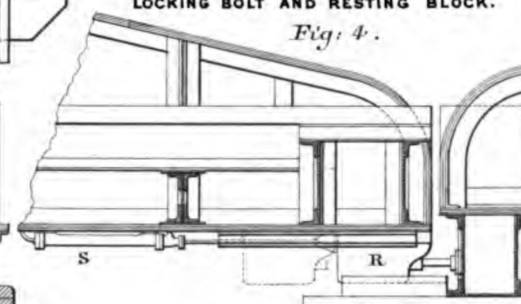
END ELEVATION SHOWING,
RESTING BLOCKS AND LIFTING GEAR.

Fig: 3.



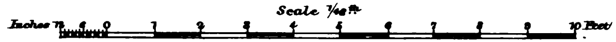
LONGITUDINAL SECTION, SHOWING
LOCKING BOLT AND RESTING BLOCK.

Fig: 4.



SIDE ELEVATION
SHOWING RESTING BLOCK.

Scale 1/48th



Thos. Kell, & Son, Lith.

3 and 4. When the toggle joint-levers, PP, are withdrawn, the bridge is lowered on the blocks. The hydraulic cylinders N and S are controlled by valves on the centre platform in reach of the bridgeman, who can stop the bridge at the right place by means of a dial with pointers actuated by the motion of the bridge. When the motion is stopped, a locking-bolt T, three inches thick (which is pressed outwards by a spiral spring) is shot out at each end of the bridge into a corresponding slot, and so locks the bridge. These blocks are withdrawn by the wire cord U when the bridge is to be swung.

The line of the bridge being north and south, a slight lateral warping is caused by the sun acting alternately on the opposite sides of the bridge. To enable the bolts to enter their slots when the warping occurs, the feet of the lifting levers, PP, are bevilled on their inner faces at II (fig. 2), and bear against corresponding bevils VV on the bed plates, by which means the ends of the bridge, when warped, are forced back into the correct line.

The accumulator being stationary, whilst the fixing gear swings with the bridge, the water-power is conveyed by a central copper pipe W (Plate 28), which passes up through the centre of the bridge, and has a swivel joint at the lower end. As the hydraulic turning-engines are also stationary, whilst the bridgeman's hand-gear rotates, the communication for working the valves is made by a central copper rod X (Plate 28), which passes down through the centre of the pressure pipe W in the axis of the bridge. The opening or closing of the bridge is accomplished in 50 seconds, the average speed of motion of the end of the bridge being 4 feet per second.

Small gas jets are provided in the central pier, and in the chambers containing the hydraulic cylinders, and are kept burning in very frosty weather. The pipes leading to the machinery at the ends of the bridge are protected by cinders encased in wooden boxes.

Another important hydraulic swing-bridge is that which crosses the river Tyne. The swinging portion of this is 280

feet long, and weighs more than 1200 tons. In this bridge, instead of the weight resting upon live rollers, an hydraulic press is applied to the centre; it has a pressure of about 900 tons upon the ram, which relieves the pressure upon the rollers to that extent. The rollers and roller path, however, are sufficiently strong to carry the whole weight of the bridge, supposing any accident were to happen to the centre press. The central press being always open to the accumulator pressure, a permanent relief is afforded without any waste of power.

Hydraulic power was first applied to drawbridges in 1853 to work the bridge over the river Tovey, on the South Wales Railway near Carmarthen; and at the Sunderland Docks an hydraulic drawbridge was constructed about the same time. In the application of hydraulic power to drawbridges, the first operation consists in lifting the bridge sufficiently high to enable it to roll back over the permanent way in the rear. The lifting presses then act directly under the main girders of the bridge, and, as the tail-end is heavier than the nose-end, the nose-end of the bridge is first raised against the roller bearings, and then, when the back-end is raised to its proper level, the bridge is hauled back by means of hydraulic reciprocating engines. These act upon rollers, either attached to the bridge itself (in which case the rollers run upon a roller path fixed to the masonry) or upon rollers attached to the head of the lifting ram, when the roller path is attached to the bridge itself. A noticeable bridge of this type has recently been constructed at the Kattendyk entrance to the Antwerp Docks.

Hydraulic power has also been applied on the "Bascule" (or old lifting drawbridge) system, both single and double leaf. A bridge of this character occurs over one of the dock entrances at Liverpool. Sometimes the dip of the "Bascule" bridge is counterbalanced by the tail-end of the bridge. In some cases the bridge is hinged on the quay level and is lifted bodily back, leaving the passage of the quay-way perfectly free and uninterrupted for passengers. There is a Bascule bridge at York in

Fig: 1.

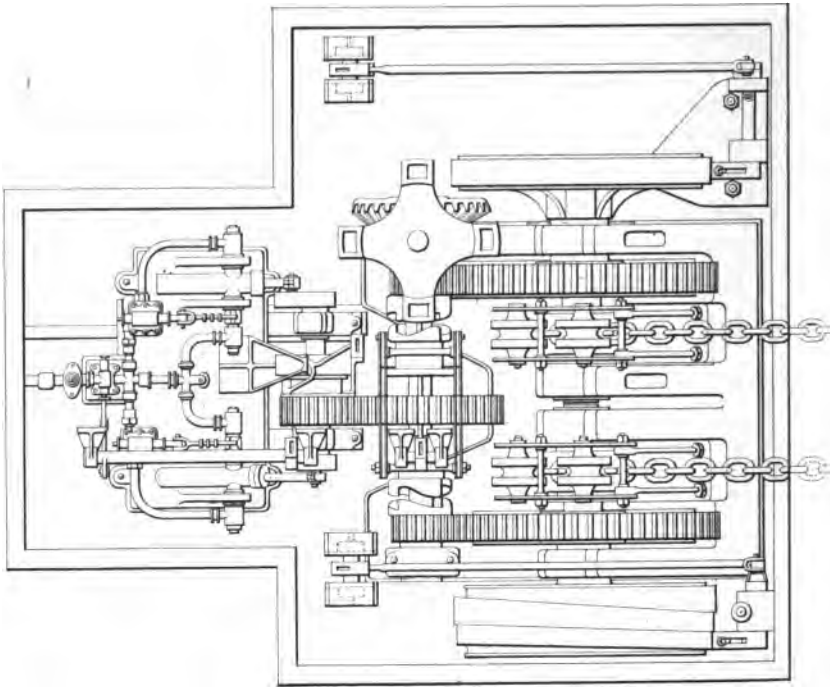
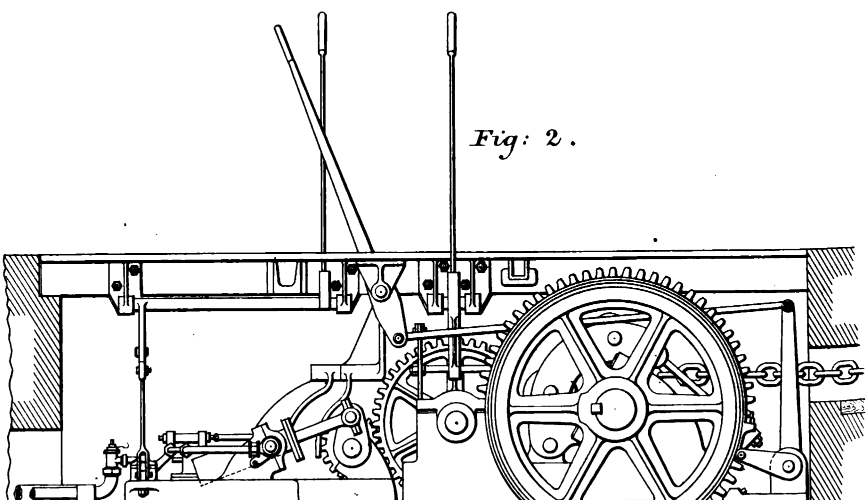


Fig: 2.



Thos Kell & Son, Lith.

one leaf of 34 feet, which is raised by chains actuated by hydraulic rams. At Copenhagen, again, there are seven Bascule bridges, one of which, having two leaves of 62 feet each, is in the very centre of the city, and has to carry a large portion of the traffic across the harbour. It is worked by hydraulic machinery, and has been opened and closed fifty-five times in a day. A Bascule bridge to cross the Thames near the Tower is now in course of construction, the first stone having been laid by the Prince of Wales in June 1886; Sir Horace Jones and Mr. J. Wolfe Barry are the engineers. The Bascule portion of the bridge, which constitutes the centre opening, is in two leaves 50 feet in width (made up with a roadway of 36 feet, and two footpaths of 7 feet each), and 200 feet span, each leaf being therefore 100 feet. The leaves of this bridge are to be raised and lowered by hydraulic machinery.

DOCK-GATE MACHINERY.

One of the first applications of hydraulic power in this direction was to the old hand-power gate machines at the docks at Newport and Swansea. Lines of shaftings were carried along the dock wall, with gearing connecting the engines to each of the hand crabs. This shafting was actuated by rotating hydraulic chains, which at the same time worked capstans at the extreme end of the dock.

Hydraulic engines were next applied directly to each crab, instead of through intermediate shafting. An "Elswick Gate Crab," with hydraulic engine attached, is shown by Plate 30, figs. 1 and 2. In this arrangement there are two rotary hydraulic engines with two cupped drums to each, a clutch enabling either drum to be thrown into gear with the engine, whilst the other overhauls. One engine with its double crab serves for two chains, which are led along the top of the gate. The chain for opening passes over a pulley, and descends vertically by the side of the gate. After passing over another

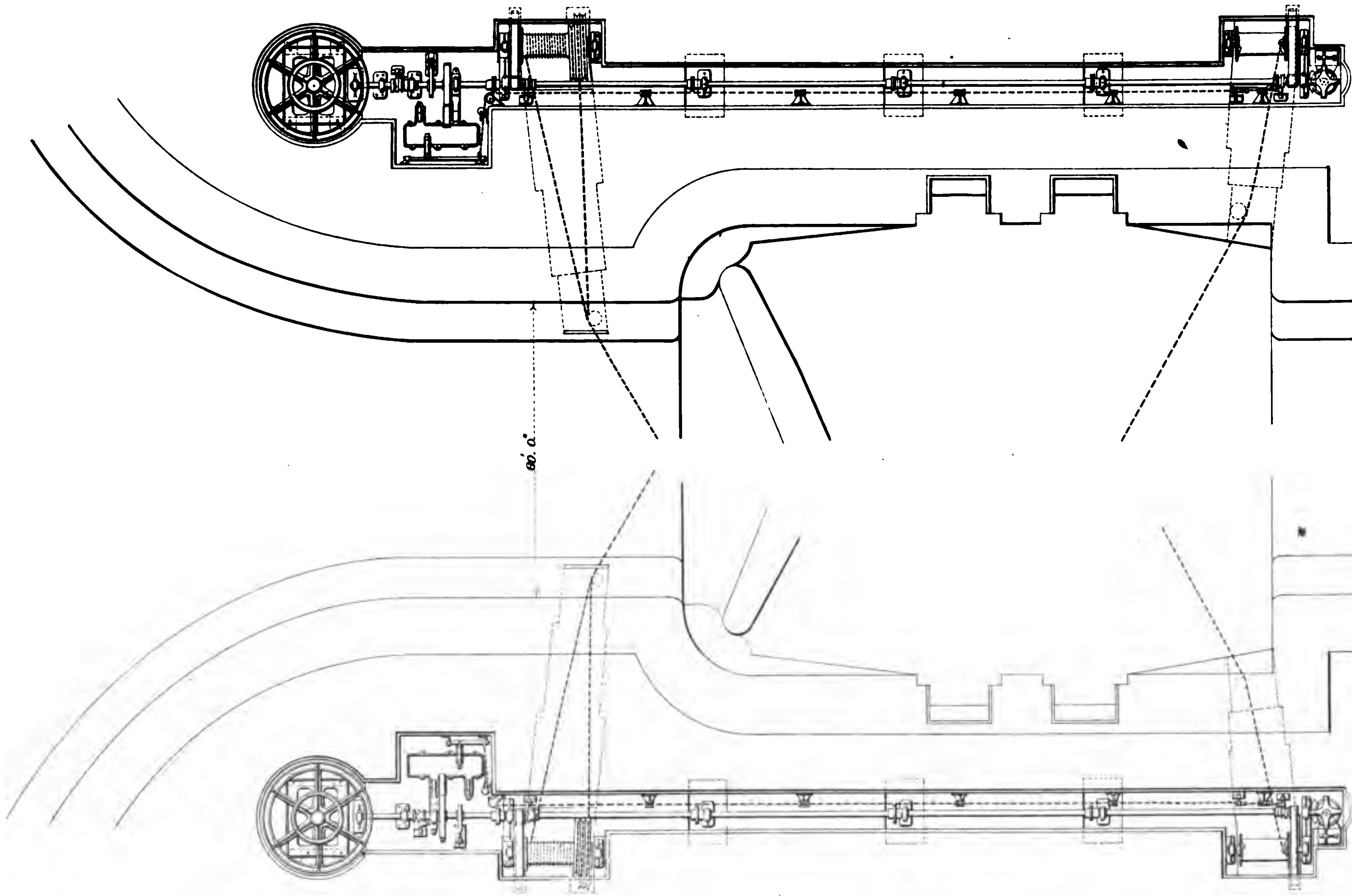
pulley, it is attached to the masonry at the place where in ordinary practice the roller-box is fixed. By this arrangement the old method of making chainways in the masonry is obviated, and each leaf of the gate is worked both ways from the same side.

Another arrangement is that by which hydraulic power is applied to the crabs through shafting driven by hydraulic engines. Plate 31 shows an "Hydraulic Capstan and Gate Opening and Closing Machinery for 60-feet Lock." By this method the opening and closing chains are led from the gates through roller-boxes in the masonry of the walls, and thence up inclined chainways to separate winding-drums. These are worked by shafting from three-cylinder hydraulic rotatory engines (one on each side of the lock). The closing drums have a spiral at the end for taking up quickly the slack chain lying across the lock. When the opening drums are hauling in, the closing drums are paying out, and *vice versâ*, to effect which the drums are connected to the shafting by clutches. The drums are controlled by brakes when paying out. The shafting can be worked by hand, by means of a sunk capstan-head, in case of need. A capstan for hauling ships is also connected to shafting by a clutch and bevel spur gearing.

Another form of gate machinery is the direct-acting ram and cylinder with multiplying sheaves, which has been very generally adopted, although it does not provide for the working of the gates by hand in case of need, or if the power is not available. Plate 32 shows an "Hydraulic Machine for Opening and Closing 80-feet Lock Gates." The opening and closing chains are led from the gates through roller-boxes in the face of the walls, and up inclined chainways, to horizontal hydraulic cylinders and rams. These have multiplying sheaves, around which the chain is led, the end of the chain being attached to the cylinders. On pressure being admitted to the opening cylinder (by means of a valve in a pit adjoining), the ram is forced out, drawing up the chain, opening the gate, and pulling

HYDRAULIC CAPSTAN

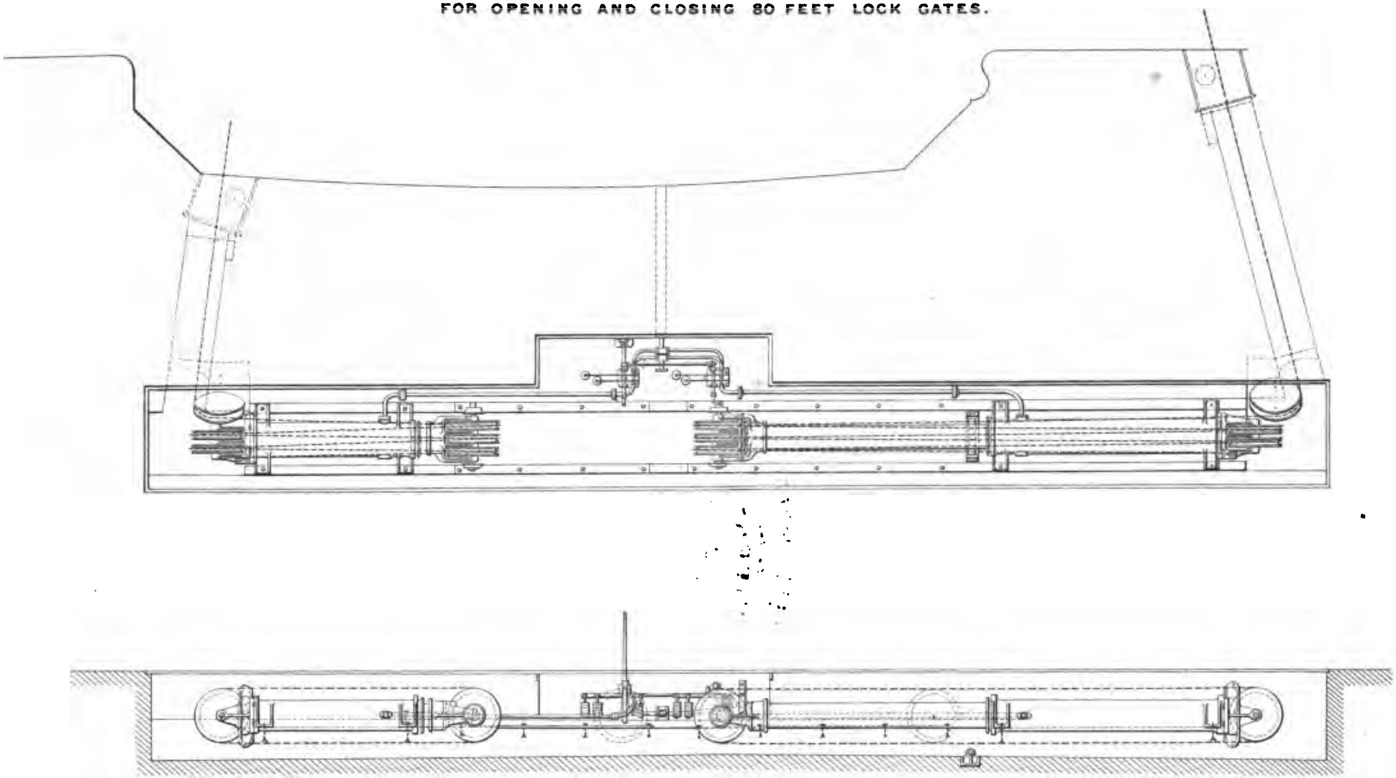
AND GATE OPENING AND CLOSING MACHINERY FOR 60 FEET LOCK.



Thos Kell & Son, Lith.

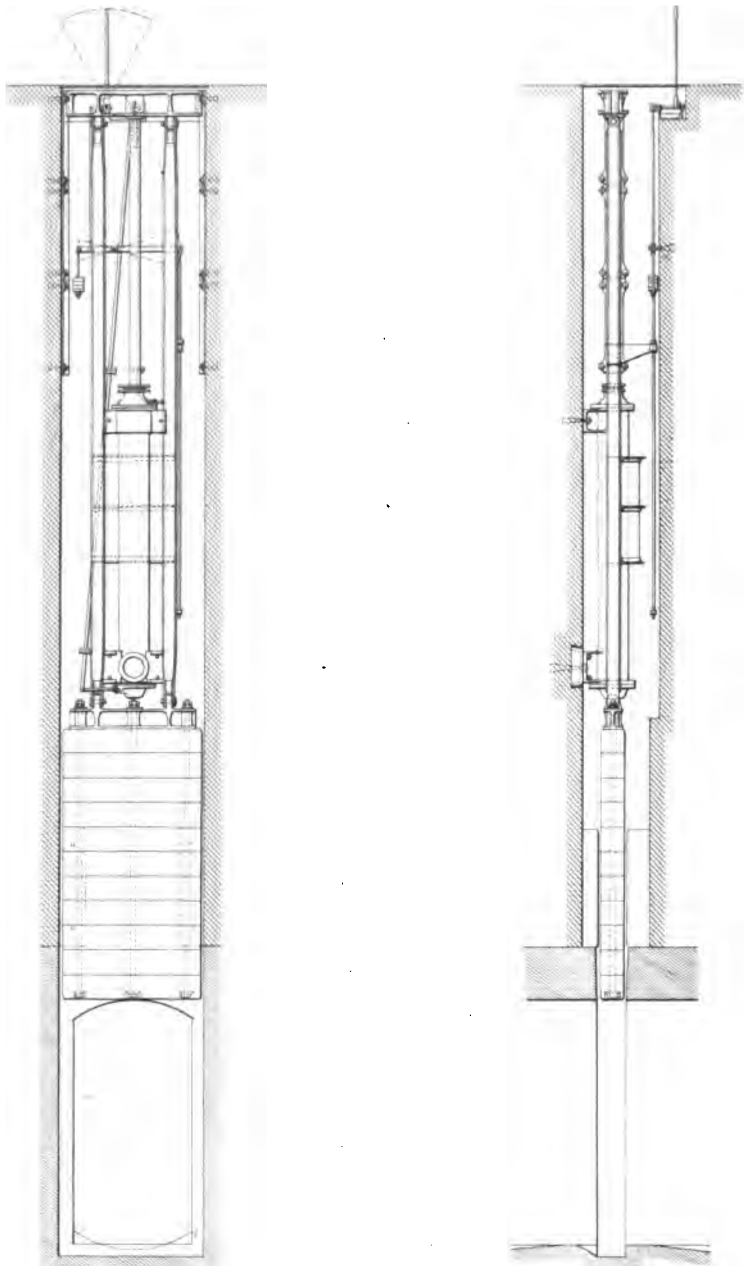
HYDRAULIC MACHINE,

FOR OPENING AND CLOSING 80 FEET LOCK GATES.



Thos Kell & Son, Lith.

DIRECT-ACTING HYDRAULIC SLUICE MACHINE.



Scale.

Inches 12 0 1 2 3 4 5 10 15 Feet

Thos Kell & Son, Lith.

in the closing ram on the other side of the lock, which (by means of its valve) has been put in connection with the exhaust. For closing the gates, the operation is reversed. The ram-heads are carried by rollers on tram-plates. This arrangement is simple, and has very few wearing parts. The strain on the chain when the gate is moved is about 10 tons. The gates are opened in a minute and a half.

The illustration only shows the machinery on one side of the lock. The other is precisely similar.

The widest entrances to which hydraulic power has been applied are 100 feet at the Canada Dock, Liverpool, at the Barrow Dock, and at Birkenhead.

Sluices for removing mud at the entrances to locks can be conveniently worked by hydraulic machinery. Their movement up and down is effected by the application of direct-acting cylinders and pistons attached to the masonry to open and close the paddle against the pressure. A hand-pump is generally applied so as to be available in the event of the water-pressure being accidentally cut off. The combination is effected by a screw and gearing worked by a rotary hydraulic engine, so arranged that it can be worked by either hand or power. The sluice cylinders are usually lined, and the rods covered, with copper. Plate 33 shows an Elswick "Direct-Acting Hydraulic Sluice Machine." At the Alexandra Docks, Newport, the hydraulic sluices are attached to the gates themselves, instead of being placed on the masonry.

Lock-sluicing arrangements at dock entrances have generally failed by reason of the scouring action of the water on the mud being limited to the mouth of the sluice. Deep holes are made there, as the scouring force is not operative at any distance from the sluice. A solid apron should therefore be carried for some distance beyond the mouth of the sluice to prevent the force of the water from disturbing the floor (as it frequently does). The current would then be directed with the best effect, and the formation of the upper eddies, which destroy the sluicing power of the water, would be avoided.

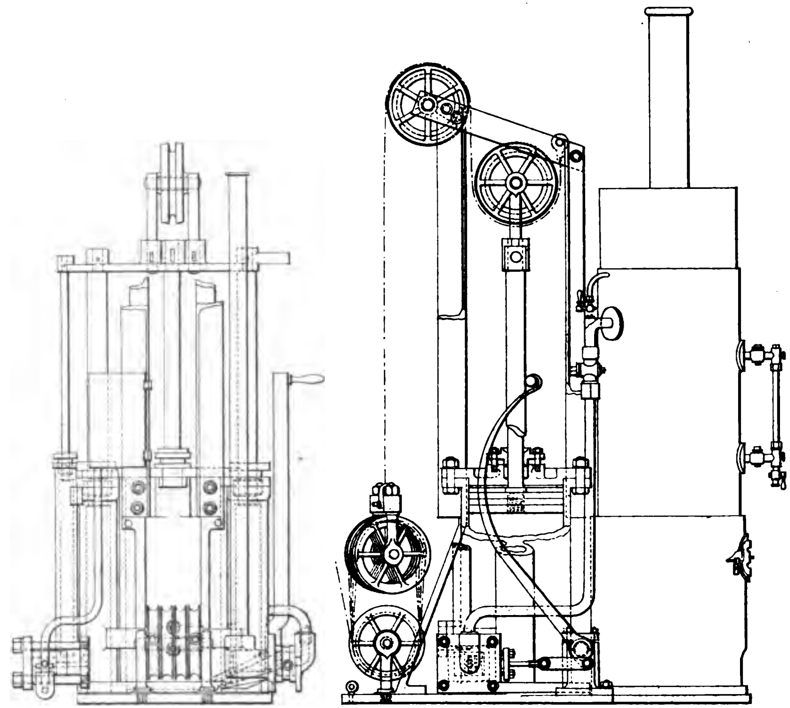
HYDRAULIC COAL-DISCHARGING MACHINES.

In the year 1851 Sir William Armstrong devised the "Coal-Whipping Machine," shown by Plate 34, for the purpose of more expeditiously discharging coal from colliers on the river Thames, in lieu of hand-whipping. The load is raised by one stroke of a steam-piston acting through multiplying sheaves on the lifting-chain. At each side of the steam cylinder there is a cataract plunger attached to the cross-head of the steam-piston, and the descent of the load is regulated by a valve, which controls the passage of the water from the cataract cylinders to a small cistern above them. A vertical multitubular boiler supplies the necessary steam.

The upper side of the piston is in constant communication with the boiler, and when it is desired to make a lift, the lower side of the piston is, by the movement of the starting-valve, opened to the exhaust. In lowering, the steam-piston is placed in equilibrium by admitting steam to the under side, and the descent of the load is controlled by the valve on the cataract cylinders. A tappet on the cross-head automatically shuts the starting and cataract valves at each end of the stroke. A feed-pump for supplying the boiler with water is also attached to the cross-head.

About the same time that the above machine was introduced on the river Thames, the Glamorganshire Canal Co. took steps to improve the method of discharging coal, which resulted in the fitting up by Sir William Armstrong of two hydraulic machines at Cardiff. A wooden frame was placed at a sufficient distance from the quay to enable a truck of coals to stand between it and the latter. The coals were discharged from the barges or vessels on the canal by a bucket, which was raised and lowered, swung inwards or outwards, by means of a vibrating jib, all the operations being performed by hydraulic power. The action of these vibrating jibs (which were afterwards used

COAL WHIPPING MACHINE.



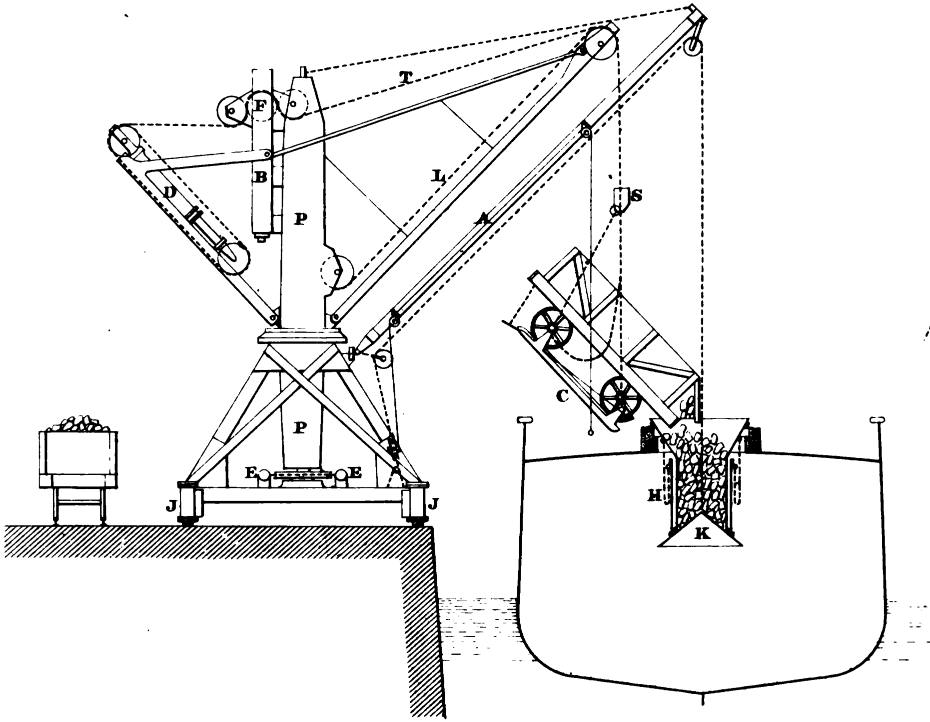
Scale.

Inches 12 9 6 3 0 1 2 3 Feet

Thos. Kell & Son, Lith.

MOVEABLE HYDRAULIC CRANE.

Fig: 1.



SIDE ELEVATION, SHOWING MODE OF TIPPING.

Scale 1/160th

Feet 10 5 0 10 20 30 40 50 Feet

Fig: 4.

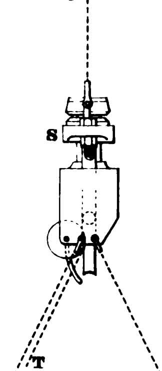
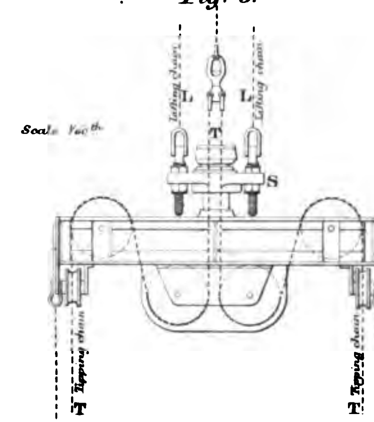


Fig: 5.



SWIVEL ATTACHMENT.

Fig: 2.

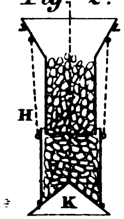
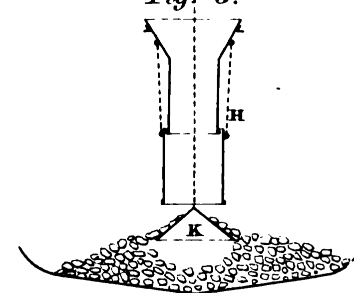


Fig: 3.



TELESCOPIC HOPPER.

Thos Kell & Son, Lith.

on a larger scale at Swansea and Poplar) was controlled at the extreme end of the stroke by a tapered rod closely fitting the aperture through which it passed.

The most recent development of the system may be exemplified by the Moveable Hydraulic Crane devised by Mr. Westmacott, and erected by Sir William Armstrong & Co., at the Bute Docks, Cardiff. A description of this was given by Mr. M'Connochie at the Cardiff meeting of the Institution of Mechanical Engineers in 1884, and is shown by Plate 35. The shipping of coal direct from the trucks had previously been carried out by fixed hydraulic cranes. It was, however, found that the work could not be done rapidly enough, as the fixed cranes could only load into one hatchway of a ship, since the positions of the hatchways in steamers varied so much that the cranes could not be placed to suit different vessels. Moveable cranes were decided upon to obviate this hindrance to rapid working; but, as the cradle or platform on which the truck was lifted required a pit in the line of rails for its reception, a crane could only pick up waggons at one point. To meet this difficulty, Mr. Westmacott designed the coaling cradle C shown by fig 1. It consists of a light platform suspended by chains capable of being placed in any position upon a line of rails. The platform is permanently hung by chains from an anti-friction swivel S (shown to a larger scale in figs. 4 and 5), which enables a man to turn the cradle with a loaded waggon on it, thereby dispensing with turntables.

The crane is carried on a nearly square wrought iron pedestal, which runs on four wheels upon a line of rails of 24 feet gauge. There are also four lifting Jacks, JJ, one at each corner, which take the weight when the crane is at work. The pillar PP consists of two flat plate girders which revolve in bearings at the top and bottom of the pedestal. There are three hydraulic cylinders for lifting and tipping; the first is placed between the girders of the pillar for lifting the load by means of the chain L, the two ends of which are made fast to the swivel attachment S. The second, D, is for tightening the tipping chains T,

and the third, B, is for effecting the tipping, by making a bight in the tipping chain (as shown at F), while the cylinder D is locked by its valves.

The pillar is turned by two horizontal hydraulic cylinders, EE (one on each side of the pillar) fixed to the pedestal, and working a chain which passes round a drum at the foot of the pillar.

All the motions are readily controlled by one man in a valve house fixed to the pedestal (not shown on the Plate). Two such houses are provided, on opposite sides of the crane, so that the man can use whichever is most convenient for watching the operations. The pressure water is conveyed to the crane by moveable jointed pipes, which can be attached to hydrants placed at convenient distances on the hydraulic mains along the quay wall.

There is an auxiliary or anti-breakage crane, A, on the side next the dock, for working a hopper, H, resting on the deck of the ship. This hopper (designed by Mr. Charles L. Hunter of the Bute Works) has a telescopic throat of square section, which is closed by a pyramidal bottom, or valve, K, held up by the auxiliary crane A. The object of this is to allow the first few truck-loads of coal to be lowered gently to the bottom of the hold, so as to lessen the breakage of the coal (as shown in figs. 1 and 3). When not in use, this crane can be swung to the side, out of the way. A waggonful of coal can be shipped in from 2½ to 3 minutes.

HYDRAULIC MACHINERY ON BOARD SHIP.

Hydraulic power is now being applied by various machines to perform the deck work on board the larger steamships, instead of the steam appliances which were previously used. In discharging a ship's cargo, the facilities for expedition now

afforded in all large docks, by the employment of hydraulic cranes and other appliances, have not until quite recently been met with corresponding facilities on board ship.

On board steamships the accumulator can be dispensed with, as the enormous reserve of motive-power in the engines themselves renders the storage purposes of the accumulator unnecessary. The power required to work hydraulic apparatus under these circumstances is generally produced, and communicated direct to the machine, by motors which are capable of working at very high speeds, and of developing in small compass a large amount of energy.

Direct-acting cylinders, working upon the single crank, have been recently applied by Sir William Armstrong, Mitchell, & Co., to ship-capstans up to five tons power. In these the bed-plate is fixed, as it would be too unwieldy to use in the form of the turn-over arrangement already described for docks and yards. The capstan-head is in some instances arranged with two diameters; the larger being for the lighter power, and the smaller for the heaviest strain upon a hawser. The capstan-head is also provided with handspikes and rack, so that, if desired, it can be worked by hand as an ordinary capstan. A separate hydraulic engine (usually with three cylinders) is attached to the masonry, and, by means of gearing, the power is conveyed from it to the capstan, the shaft of which is firmly fixed to the foundation, the upper part being carried by cast iron framework.

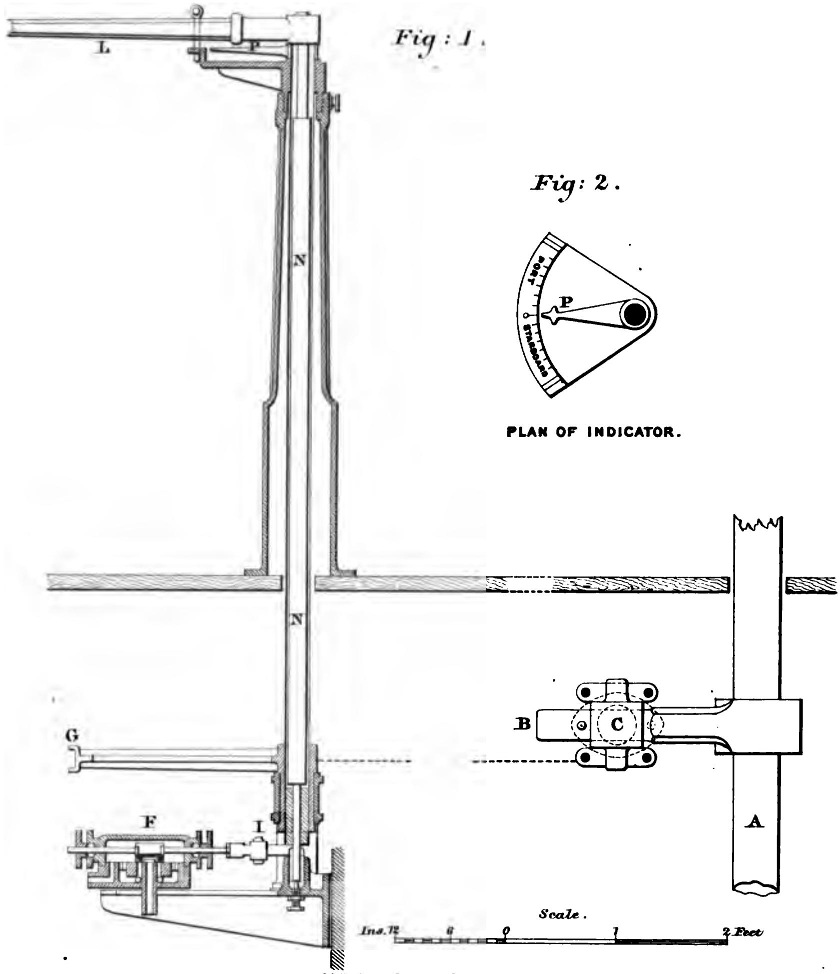
The greatest power that has hitherto been given to ships' capstans is represented by a pull of 11 tons on the hawser, although, as a rule, a 5-ton capstan is found sufficient. A variation of power may be given either by means of gearing, or by applying a triple power hydraulic capstan engine either direct to the capstan or through gearing. In practice, however, it is generally found that the saving of power effected is not commensurate with the complication involved.

Mr. A. Betts Brown has devoted much skill and attention to this branch of hydraulic machinery. In papers communicated to

the Institution of Mechanical Engineers, and to the Institution of Naval Architects, he has given data as to the employment of hydraulic apparatus of various kinds on board the *Quetta* (of the British India Steamship Company). The prime mover for the hydraulic machines on this ship consists of a pair of compound surface-condensing pumping-engines of 100 indicated horse-power, independent of the ship's engines. The average speed is 40 revolutions per minute (never exceeding 70 revolutions). The engines are connected with pumps attached to a steam accumulator, with a pressure of steam from the steam-pipe of 80 lbs. to the square inch. This pressure acts on a steam piston and water ram, having areas in the proportion of 10 to 1. A pressure of 800 lbs. to the square inch is thus produced in the hydraulic system.

On board the *Quetta* two single lifts are placed at the extreme hatches, and two pairs at the main hatches. These consist of long cylinders (passing through the upper and main decks), which contain lifting rams, each having three sheaves, a corresponding number being fixed in the upper deck. By this arrangement a lift of 70 feet is obtained. The weight lifted by each is 30 cwt., but, by connecting a pair together, 3 tons can be raised. The speed of lifting averages 5 feet per second, and to prevent accidents at this high speed (in the event of the cargo breaking away) an automatic arrangement is provided. An appliance called the "Hydraulic Derrick Topping Gear" is attached to the mast. It consists of a cylinder having a ram, with one sheave carried at its end, working downwards, so that its weight may tend to balance the weight of the derrick. The fast end of the chain is held by a clip; it passes round the sheave on the ram, thence over a swivel-pulley on the mast, and lays hold of the end of the derrick. A small slide-valve is placed close to the hoist and hatch, the admission port of which is connected to the cylinder by a small pipe up the mast. The hoists are sometimes arranged in pairs (where there are two derricks), with a communicating pipe and valve.

Fig: 1.

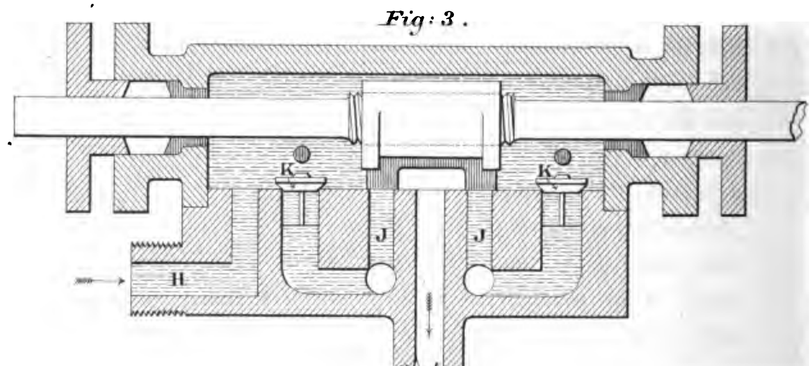


VERTICAL SECTION.

Fig: 2.

PLAN OF INDICATOR.

Fig: 3.



SLIDE VALVE.

Scale.

In. 1 ½ 0 1 2 3 4 5 6 Inches.

Thos Kell & Son, Lith.

The positions of the fore and aft hatches of this ship with reference to the masts were such that while the derricks were set to plumb the hatches, they would not reach over the side of the ship on swinging round. This difficulty was overcome by attaching to the mast a small hydraulic cylinder, with its ram working downwards, carrying a sheave round which the derrick-chain passed. From this cylinder (which is placed some twenty feet up the mast) a small pipe is led to a valve placed close to the hatchway. The man in charge works the hoist lever with his right hand, and with his left the derrick lever.

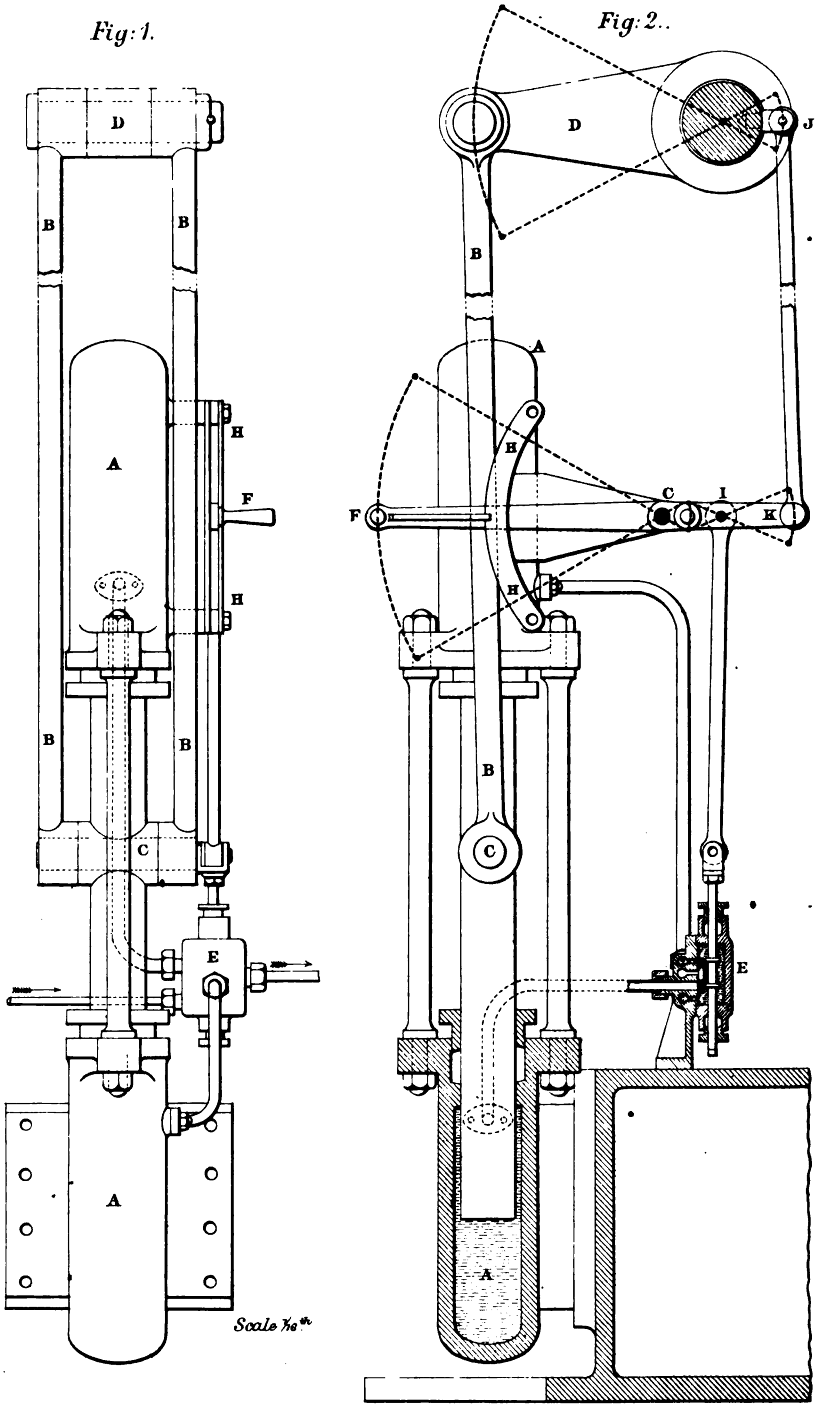
Hydraulic power is also employed to work the steering-gear of ships. It is necessary that the power applied should increase in proportion to the angle at which the rudder is moved over, and that the machinery should yield under any excessive strain, so as to allow the rudder to fly amidships, but to return after the strain has passed away. The valve for controlling the steering gear should be placed where it can be worked under the eye of the officer of the watch. Mr. Betts Brown has arranged hydraulic steering-gear as shown on Plate 36.

A is the rudder post, B is the main tiller, which is keyed to the post. The end of the tiller is cylindrical, so as to allow the sliding-block C to slide radially upon it. This block is connected by trunnions to hydraulic rams, working in a cylinder, from which separate pipes are carried to the admission ports in the slide-valve F, which is placed at the bridge. When the main tiller is moved in either direction towards its extreme position, the sliding-block runs out upon it, and the proportionate extent of motion in the effective leverage of the rams is increased until the power over the rudder becomes doubled, when the rudder is hard over at 45°, on either side of the midships position. A wire cord passes from the rams to a quadrant G, by which the motion of the rudder-post is communicated to the quadrant. This forms an automatic cut-off by the slide-valve F, and serves also as a deck indicator. The steering valve F is shown in section to a larger scale in fig. 3. It is three-ported, the two end-ports JJ leading

through the pipes to the hydraulic cylinders astern; the centre port being the exhaust. The water enters the valve-chest by the port H. Relief-valves KK provide for the shocks caused by a heavy sea striking the rudder. The steering tiller L (fig. 1) is fixed upon the shaft N, which passes down from the bridge to the valve F on the maindeck, and terminates in a crank at the bottom. This crank works one end of a floating lever I (fig. 1) by a pin and connecting-link. The other end is attached by a similar connecting-link to a crank-pin in the quadrant G. To the middle of this lever I is joined the slide-valve spindle. If the valve is opened by moving the steering tiller L and the lever I, water is at once admitted to one of the steering cylinders and is exhausted from the other, causing the ram to move the rudder. But as the quadrant G receives motion from the rams by the wire cord, and therefore moves through precisely the same angles as the rudder, its crank-pin is carried in the opposite direction to the crank on the shaft N, and the slide-valve is by that means shut. Any further movement of the steering tiller L will produce further motion in the rams, with a corresponding counteracting motion of the quadrant. The slide-valve is also opened immediately by the quadrant whenever the rudder is driven amidships by a heavy sea, and thus a double relief is afforded to the water. The quadrant closes the valve again on the rudder returning to the position from which it was disturbed. The shaft N of the steering tiller has a tubular casing which is fixed to the quadrant below, and is carried to the top of the steering pedestal with a pointer P (keyed on it), to indicate by a graduated arc the position of the rudder as shown by fig. 2. In the steering-gear for ships of the *Quetta* type, large relief-valves are placed on each cylinder. Their exit pipes cross over to each other, so that when a sea strikes the rudder, the water is forced out of the cylinder acting against it into the opposite one. These valves are set to blow off when the rudder is subjected to over one-fifth of its breaking strain.

Plate 37 shows Mr. Betts Brown's "Hydraulic Reversing

HYDRAULIC REVERSING GEAR.



Thoˢ Kell & Son, Lith

Gear" as applied to the *Mikado* (3000 tons). The apparatus consists of two hydraulic single cylinder engines AA (figs. 1 and 2), with rams 4¾ inches diameter and 19-inch stroke, coupled together and working in opposite directions. They are connected by slide-rods B, from the boss C to the weigh-shaft lever D. Water is admitted to either of the cylinders by opening the slide-valve E by the handle F, which is centred at G, and has a detent rod and quadrant H. The reversing handle F is connected to one end of the short double lever K, the other end of which is moved by a connecting-rod from a stud point J on the back of the weigh-shaft of the main engines. The slide-valve spindle is attached to the double lever K at an intermediate point. When water is admitted to the lower cylinder by a downward movement of the reversing handle, the slide-valve is raised, and the hydraulic rams are moved in an upward direction, carrying with them the weigh-shaft lever D, and reversing the engines. By the same movement the stud point J upon the weigh-shaft is lowered, and closes the valve again, so that the two rams are held fast in whatever position they may be placed. This counteracting cut-off enables the engineer at once to place the reversing links of the main engines at any degree of expansion, and to hold them there, by simply moving the reversing lever into the desired position in its quadrant. The rams follow at once into that position and stop there.

HYDRAULIC PILE DRIVER.

In the construction of the Alexandra Docks, Hull, Messrs. Lucas & Aird employed an hydraulic hoisting machine for the purpose of driving piles. The machine thus utilised consisted of an ordinary grain hoist with the working chain-drum removed. The monkey chain was worked direct from a ram

7⅝ inches diameter, and 3-feet stroke. A recessed chain sheave, with grip gear fitted to it, acted as a brake, and prevented the chain which leads to the monkey from slackening. Intermediate sheaves guided the chain from the ram to the monkey. Another sheave took the chain (after it had passed over the ordinary ram sheave), and to the shaft of it a brake wheel was keyed, which was used to prevent the chain from slackening out in the direction of the balance weight. The monkey chain was led over the top of the pile engine in the usual manner.

Were it only required to work the monkey at a stated height, the brake appliances and the balance weight would be unnecessary, as the ram would do this with the end of the chain made fast as usual. Where, however, the height at which the monkey had to be worked varied through the level of the ground from 40 feet to 45 feet, it could not be done without some such arrangement for shortening up, or letting out, the chain. This was also required for lifting the pile from the ground for fixing.

Knuckle-jointed pipes connected the machine with the hydraulic main, and could be doubled up or extended, as the machine moved in either direction.

The action of the machine will perhaps be best shown by describing the lifting of a pile. The chain is attached to one end of a pile on the ground. The brake is applied to prevent any movement at this end of the chain. The ram is set in motion, and having a 3-feet stroke, with multiplying power 4 to 1, its pullies lift the pile about 12 feet. The grip gear is then applied to prevent the pile from lowering, as the ram recedes. The brake is then loosened, and the ram is allowed to recede, the slack chain being taken up by the balance weight which lowers. When the ram is fully back the brake is again applied, and the pressure is turned on the ram. The grip gear is loosened, and another similar lift is taken with the pile. In driving the pile, the monkey is lifted to the required height in a similar manner to that described. The

brake is then applied, and is kept constantly on, until it is desired to work the monkey at a lesser height. The ram then works the monkey at whatever stroke is required. When the monkey has to be lowered (as the pile is being driven) the grip gear is applied when the ram is back, the brake is loosened, and the ram, being set in motion, lifts the balance weight, taking the chain from this end. When a sufficient quantity of chain is obtained, the brake is again applied and the grip gear loosened. Then as the ram recedes the monkey is lowered.

This machine was found to work very smoothly, and it fully answered the purpose for which it was employed. It gave fourteen blows per minute through an 8-feet drop, with a monkey weighing one ton. This rate of working compared favourably with other pile drivers which were at work at these docks, as the maximum number of blows per minute with a similar drop was only eleven.

HYDRAULIC EXCAVATOR.

During the execution of the works at the new Alexandra Dock, Hull, it occurred to Mr. John Aird to excavate the material by utilising the water-pressure in the mains that were laid for working the permanent hydraulic machinery of the dock. A machine to accomplish this was accordingly constructed, as shown by Plate 38. The circumstances were favourable to the utilisation of hydraulic power, inasmuch as it was within easy reach of the machines to which it had to be applied. The "Hydraulic Navvies" (as they were called) were, however, at times working at a distance of half a mile from the source of the power, and as there was several cranes, hoists, and other machines which abstracted power at various intermediate points, it was considered that the effective pressure at the

"Navvies" was about 700 lbs. per square inch, although the accumulator pressure was 750.

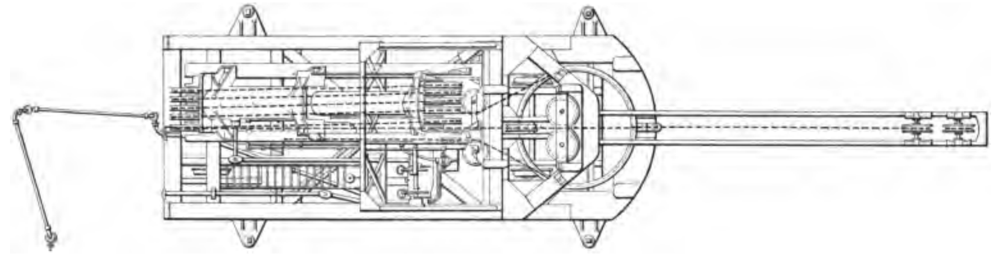
The machines are fitted with main rams, E, 14 inches diameter, and 4 feet 6 inches stroke. The chain for drawing one scoop through the excavation works over sheaves, F, the multiplying power being 10 to 1. The chain at the scoop end is worked over sheaves, G, twofold, thus giving a ratio of 5 to 1 in speed and stroke on the ram and bucket. The range generally required for the scoop was from 15 to 18 feet. In ordinary working about 3 feet 6 inches stroke was required on the ram. There are two smaller rams H (to slew the main jib) 4½ inches diameter, and 4 feet 2 inches stroke, which are fixed horizontally on the top of the bottom framing. The chains from these are attached to opposite sides of the circular platform at the bottom of the jib.

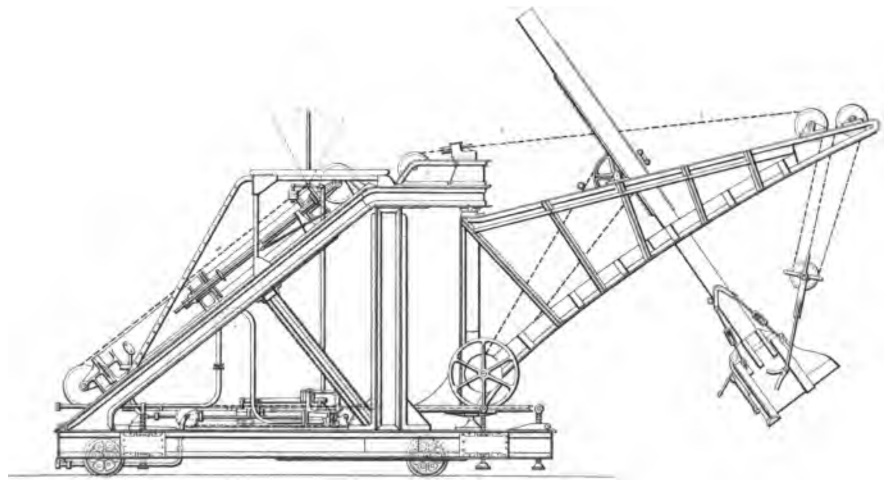
The machines are moved backward and forward by means of an hydraulic cylinder J, 3½ inches diameter, fitted with a piston. The piston-rod is attached to a rocking lever K, about 2 feet 6 inches long, which is centred on the leading axle. This lever is fitted with two catches, reversed, which gear into a double reversed toothed ratchet-wheel M, on the same axle L. The cylinder is arranged so that the pressure can be applied to either side of the piston, therefore by putting either one or other of the catches into gear, the machine is moved backward or forward. The stroke of the piston is 10 inches, and the ratchet gearing is so arranged that the machines travel about 4 inches each stroke.

The machine being set to work in a cutting (say) from 15 to 18 feet deep, the scoop is drawn up by the main ram through the "face" of the excavation, taking a cut from 4 inches to 6 inches thick, which is just sufficient to fill it by the time it reaches the top. The jib is then slewed round (by the 4½-inch horizontal rams), when the scoop is brought directly over the waggons on either side. The catch which holds the door at the bottom of the scoop is then freed, and the load is discharged into the waggon. The jib is then slewed the reverse way, and

HYDRAULIC EXCAVATOR.

PLATE. 38.





Thos Kell & Son, Lith.

the scoop is lowered, by exhausting the water in the main cylinder. The scoop weighs about 25 cwt., and has a capacity of 1½ cube yard, making a total dead load of about 3¼ tons, independent of the resistance due to the scoop cutting through the material.

The hydraulic pipes close to the machine were in 9-feet lengths, fitted with knuckle-joints, which admitted of their being so connected to the main pipes that they could be doubled up at starting, and could be extended as the machine advanced for a distance of about 18 feet, before additional pipes were required for the main. When this distance had been reached, the knuckle-jointed lengths were again brought forward and doubled up.

The greatest quantity of material that one of these machines excavated in a day was about 750 cube yards. The circumstances, however, under which they were worked were far from favourable, as the material was of a soft slimy nature, and caused difficulty and loss of time in keeping the machines in a level position. Under more favourable circumstances, these machines were considered to be capable of excavating 1000 cube yards per day each.

It was found that the "Hydraulic Navvy" had an advantage over the "Steam Navvy," owing to there being fewer wearing parts. The action of the working parts was also much smoother, by which the vibration was reduced, and fewer repairs were necessary. A saving resulted, not only in the cost of repairs, but also in the time for doing them.

HYDRAULIC DRILL.

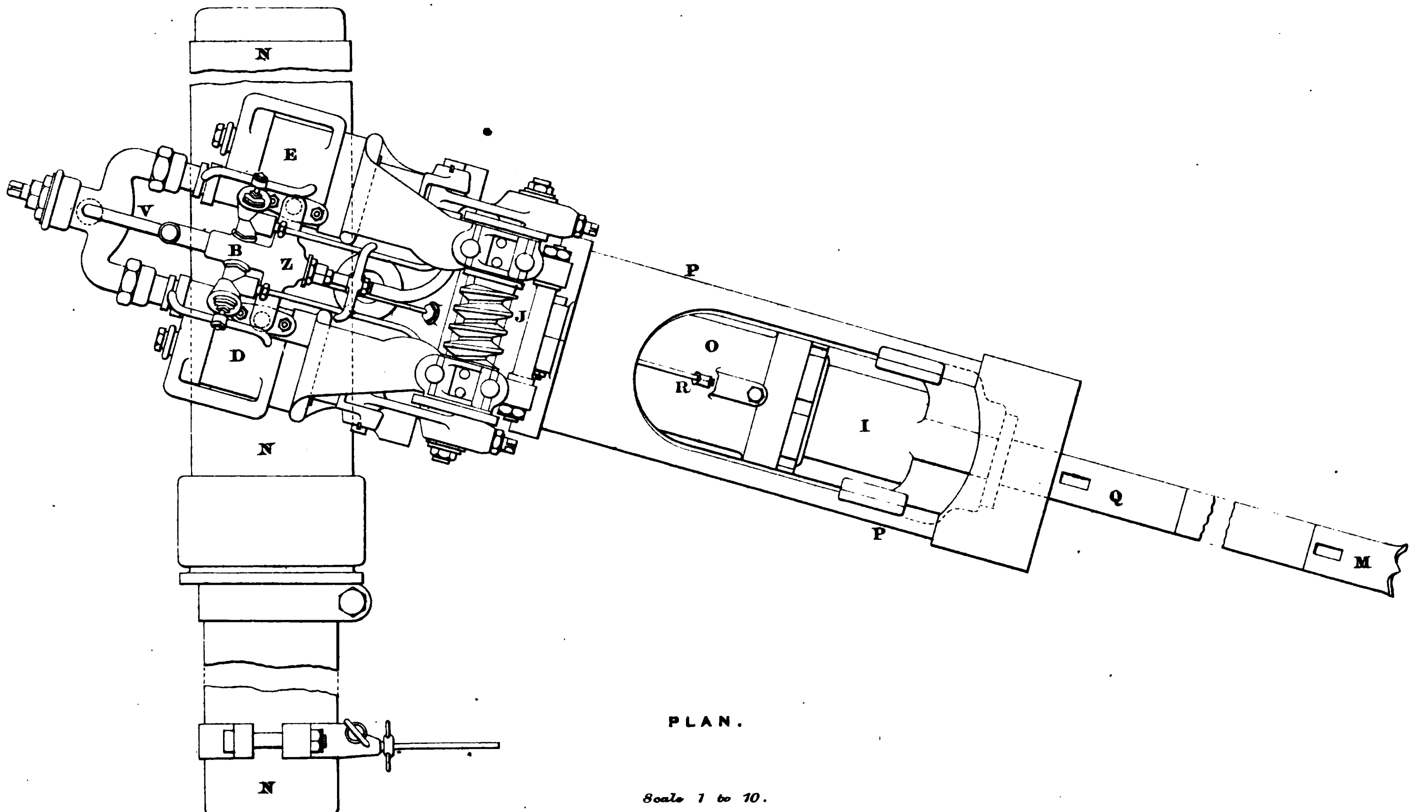
In the construction of the St. Gothard Tunnel, hydraulic power was successfully employed for the purpose of rock-drilling. The Brandt rotary drill was used at the Pfaffensprung Tunnel on that railway, and more recently at the eastern end of the

Arlberg Tunnel. The power was obtained from two high-pressure pumps, which were worked by a turbine pumping into an accumulator at a pressure of 1200 to 1500 lbs. per square inch. A $1\frac{1}{2}$-inch wrought iron pipe conveyed the water to the machine. Plates 39 and 40 show the construction of the drill, as explained to the Institution of Mechanical Engineers. The drill M is hollow, and is screwed on to the hollow bar Q, which is attached to the plunger I of the ram O, working in the guide cylinder P. Upon this guide (and in one piece with it) is a spur wheel H driven by the worm J. The whole machine is moveable from the horizontal tube N (to which it is attached) by the collar piece F. The stop cock Z in the valve-chest B admits the water into a branch pipe leading to the motors DE (about 13 HP each). These drive the worm J, which rotates the drill, at the rate of from 7 to 10 revolutions per minute. By opening the tap Y the water is admitted through the pipe U to the back of the drill plunger I, thereby pressing the drill against the rock with a force of from 10 to 12 tons. When the tap Y is closed, the water passing through the pipe R drives back the plunger. The extent to which the tap Y is opened regulates the pressure upon the plunger.

The hole can be washed out, to clear it of débris, by closing the cock C in the pipe leading from the motors to the escape hose S. The exhaust water from the engines then passes by the pipe V into a pipe in the cylinder O, and is discharged through the hollow plunger and drill into the drill hole.

The supporting pillar N consists of a tube with a plunger fitted into it. By admitting water-pressure into this tube, the plunger head is forced out against the sides of the heading, by which the pillar is set fast. The plunger can be withdrawn by means of a two-way cock. The pillar and drills are carried on a trolley, and are counterbalanced so as to be in equilibrium when the pillar is not fixed.

BRANDT DRILL,

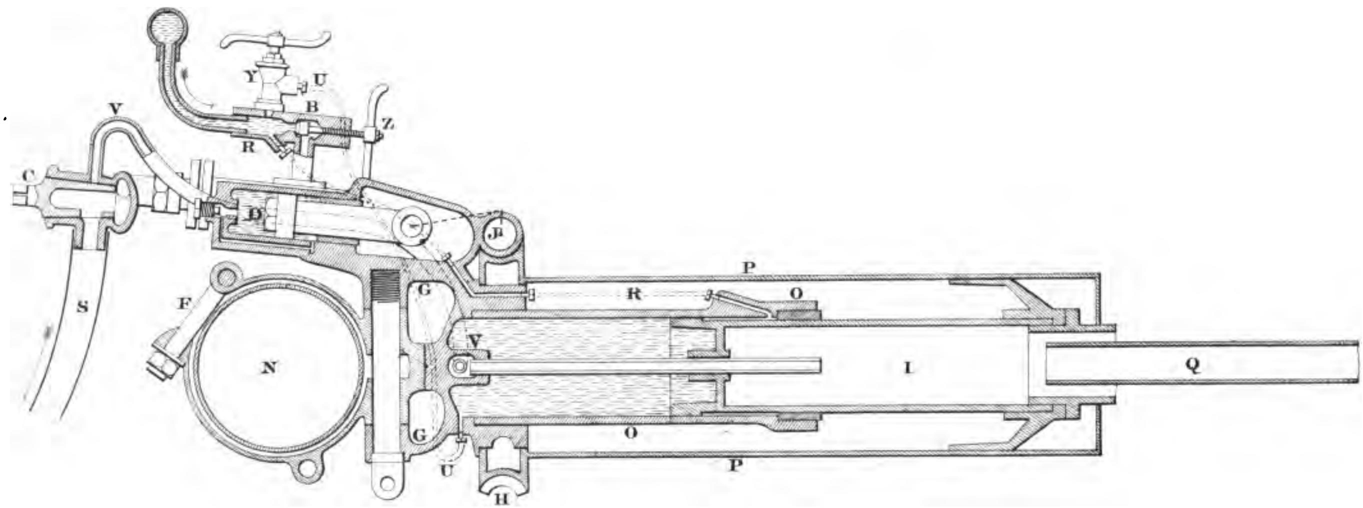


PLAN.

Scale 1 to 10.

Thos Kell & Son, Lith.

BRANDT DRILL.



LONGITUDINAL SECTION.

Scale 1 to 10

Thoˢ Kell & Son, Lith.

Fig: 1.

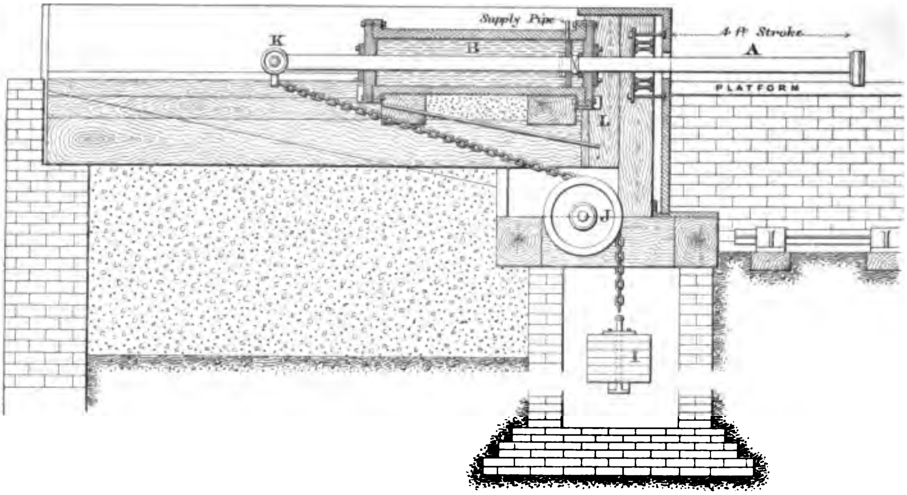


Fig: 2.

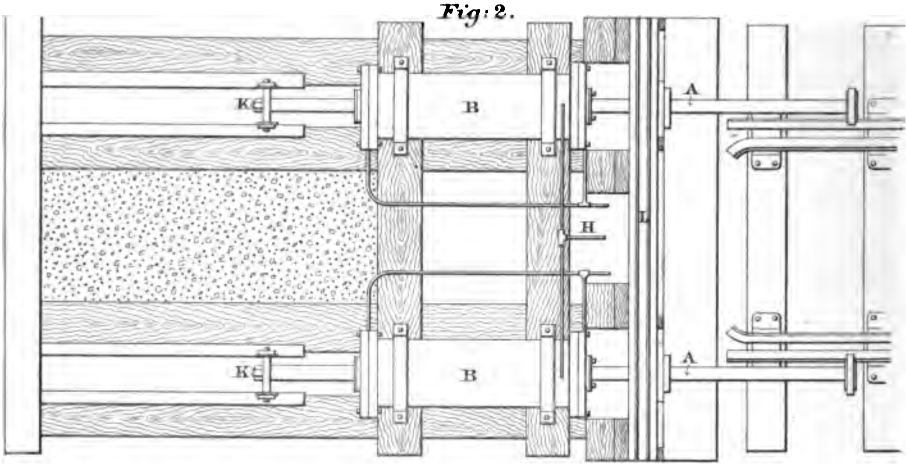


Fig: 3.

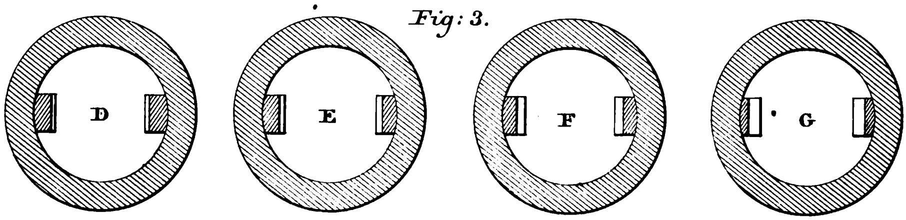


Fig: 4.

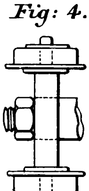
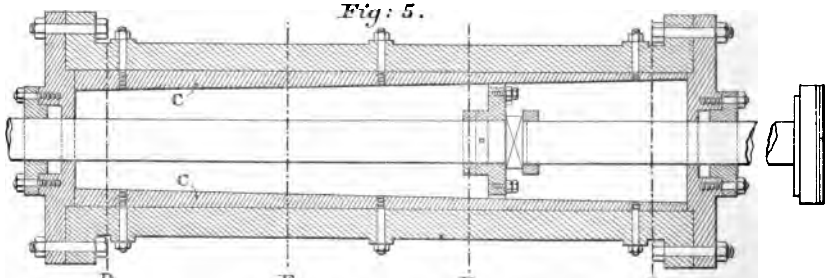


Fig: 5.



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HYDRAULIC BRAKE.

A brake or buffer can be made by enclosing a volume of water in a cylinder that is provided either with a perforated piston or with a small pipe connecting each end of the cylinder, which admits the passage of water slowly from one end to the other. Such an appliance admits of wide application in arrangements of machinery where the absorption of suddenly arrested energy is required. In working the starting and reversing gear of marine engines, the piston-rod of the steam cylinder is continued and forms the piston-rod of a hydraulic brake cylinder, the ends of which are connected by a small pipe through which the fluid is pressed backwards and forwards. When the piston begins to move, the resistance of the brake is at a minimum, increasing with the motion of the piston (as the square of the speed) till a maximum is reached, which is adjusted by means of a cock.

Mr. Alfred A. Langley of Derby has devised an "Hydraulic Buffer Stop" for the purpose of preventing accidents due to trains overrunning terminal stations, or dead ends, on railways, and arising either from the failure of the brakes, defects in the machinery, or carelessness in not reducing speed. A description of this was given to the Institution of Mechanical Engineers in 1886.

The principle on which it is based is the application of hydraulic resistance by the use of a piston working in a horizontal cylinder filled with water, and fixed in line with the buffers of the rolling stock. Plate 41 illustrates this appliance. Figs. 1 and 2 show a sectional elevation and plan of the general working arrangements. The piston-rod A working in the cylinder B is of solid steel, $3\frac{3}{4}$ inches in diameter, and 13 feet 1 inch long over all. Upon its extremity is fixed a buffer head, similar to those of the rolling stock. In its normal position (ready to receive a train) it projects six feet from the face of the cylinder, allowing two feet for the construction of a fixed stop, as shown at L. This consists of four permanent-way rails placed

transversely across the front end of the cylinders, two over and two under the piston-rod, and connected together by a loose girder through which the rod passes. The cylinder B is shown in sectional plan by fig. 5. It is 4 feet 7½ inches long, cast with a flange on each end, and bored out to 12 inches diameter, with 2½ inches thickness of metal. Covers are bolted to both flanges, and are fitted with hydraulic glands, with cup leather packing, for the piston-rod, which passes through both ends of the cylinder. An india-rubber ring 1 inch thick is fixed round the rod on each side of the piston, to form a cushion between the piston and the cylinder ends, the piston being turned to an easy fit. The constant circumferential clearance between the cylinder and the piston is 0·38 of a square inch. In addition to this constant space, a gradually diminishing area of passage has been contrived, whereby a uniform resistance is maintained throughout the stroke. This is accomplished by a wrought iron strip C, three inches wide, fastened by stud-screws along each inner side of the cylinder. They project $\frac{9}{16}$ of an inch into the cylinder at the commencement of the stroke, and taper up to $1\frac{11}{16}$ inch at the rear end. This wrought iron strip fits into a corresponding slot $1\frac{11}{16}$ inch deep, which is cut out in each side of the piston.

When an impact takes place, the piston is forced backwards. The clear space between the tapering strips and the slots in the piston becomes less and less, as shown by the diminishing areas of the waterway (the thin rectangular strip with the hatching) in sections GFE and D, so that, notwithstanding the diminishing speed, an equal amount of resistance is maintained until the train is at rest. The waterway of G is 4·96 square inches, of F is 3·18 square inches, of E is 1·40 square inch, and of D is 0·08 of a square inch. By means of adjusting screws, applied to gauge plates, the proper sizes of the openings through the pistons were determined by experiment. The cylinder is kept constantly filled with water by a supply pipe H fixed to the front of it. When released, the piston is drawn forward again into its original position by the action of a counter-

weight I, suspended in a pit under the forepart of the buffer by a $\frac{3}{4}$-inch chain, which passes over a fixed pulley J under the cylinder, and is attached to a cross-head K upon the back end of the piston-rod. This cross-head has a wheel on each side (as shown in detail in fig. 4) running along a guide path. The counterweight is composed of cast iron discs, with a packing of felt between each, and a packing of india-rubber between the bottom weight and the holding bolt, to take the first strain upon the chain, when the buffer is struck. Each buffer is designed to work separately, in order to avoid the unequal compression which might occur if two were connected by a cross-head.

It has been found by experiment that a train having a speed of at least eight miles an hour is brought to a stand with less than a 4-feet stroke. The theory of the action of the stop is considered to be, that its resistance varies as the square of the velocity of the train, while the momentum of the train also varies as the square of its velocity, so that the piston-rods will be forced back, on impact taking place, through the same actual stroke (approximately), whatever be the velocity.

On board men-of-war, a "Compressor" or "Service Buffer" is employed to check the recoil of the guns. In this apparatus the piston of the brake is perforated, and the piston-rod passes out at one end only, and is attached to the gun-carriage. By this the recoil of the gun on being fired is checked, and at the same time the energy produced by it is utilised.

HYDRAULIC GUN-CARRIAGES.

Several applications of hydraulic power to working guns have been brought before the Institution of Civil Engineers by Mr. G. W. Rendel. He described the "Hydraulic Recoil Press," which was made at the Elswick Works for the gunboat *Hydra*. This press was arranged to perform the double purpose of check-

ing recoil, and of moving the gun in or out along the slide, the recoiling gun driving back the piston. The gun was supported on two trunnions, and upon a saddle under the breach the trunnion arms rest in two sliding-blocks, which run in guides on fixed beams built on the floor of the turret. The hydraulic recoil presses were placed behind the trunnions for running the gun in and out, and were so arranged as to act simultaneously. The saddle travelled along a curved beam hinged at the back, and was raised to any required position by an hydraulic press in front of it. The cleaning and loading of the gun was also performed by hydraulic power, several systems having been devised for that purpose. One consisted of an hydraulic tube rammer, in which the head formed a sponge for cleaning the bore. It also contained a self-acting valve which opened when it was pushed against the end of the bore, and which discharged a strong jet of water within the gun. The rammer head was so arranged that this valve did not come in contact with the shot when ramming it home, nor did the valve open in ramming home the cartridge, the resistance being sufficient to bear the pressure against the bag without yielding. By these appliances two men controlled all the movements of a pair of the heaviest guns, and loaded and fired them without other help than that required to bring up the ammunition. In applying hydraulic power to these purposes, the water was pumped direct into the pipes without passing through an accumulator.

For land defences, hydraulic and hydro-pneumatic gun-carriages have been constructed for raising and lowering guns, so that the loading can be done at a low level (under the protection of a fortification), and the gun can then be raised above the parapet to be fired.

Major Moncrieff has utilised the power which is developed in the recoil of a gun, by storing it for raising the gun. In the Moncrieff hydro-pneumatic gun-carriage, the recoil of the gun, mounted on the "disappearing" principle, is taken by a combination of water and air. When air alone is employed, a practical difficulty exists in expelling the whole of it from the

recoil cylinder. The small volume of air that remains is under greater pressure than is necessary for balancing the weight of the gun, and this produces a partial recoil. The arrangement which obviates this, consists of a water cylinder and air vessel, connected by a passage fitted with an automatic valve, which opens from the cylinder towards the air vessel. The force of the recoil is transmitted through the ram in the water cylinder to the elastic medium of the air in the air vessel. This, however, cannot react upon the water owing to the intervention of the valve. The elasticity of the air becomes an agent for storing the energy of recoil for utilisation in working guns on this principle. This is accomplished by opening a communication between the water in the air vessel and the hydraulic cylinder, by which the air-pressure forces the gun upwards into position for training. By these arrangements a gun is raised into position by the power which has been derived and stored during firing. After the gun has been fired it retires of itself, and the force of the recoil is absorbed and stored. The construction of the Moncrieff Hydro-Pneumatic Gun Carriage has been chiefly carried out by Messrs. Easton & Anderson, although Sir William Armstrong, Mitchell, & Co., have applied a hydro-pneumatic disappearing carriage to Armstrong guns.

JETS.

When a jet of water issuing from a vessel passes through a short cylindrical pipe, so that the particles of water flow parallel to the axis of the pipe, and fill it, the discharge is calculable (as already explained at page 5) by taking the velocity at the extremity of the cylinder at ·82 of that due to the head (the head due to that velocity is only ·67 of the actual head, the head varying as the square of ·82). The increase in the coefficient is attributable to the fact that the side of the pipe, or

cylinder, directing the jet, attracts and retards the particles touching it, and when the vein issues and fills the pipe, an increased velocity exists in the contracted vein, with a consequent diminution of pressure. When the short pipe through which the jet passes is contracted slightly at the extremity, the discharge is increased. The first contraction at the point where the water enters the cylinder causes a reduction of velocity due to head, the second contraction at the point of exit from the tube causes the section of the vein of water to be slightly less than that of the nozzel, but increases the velocity and the discharge.

For instance, in fig. 37, water issues from a vessel through a trumpet-shaped mouthpiece A. The flow of water is contracted at BB, but where it issues to the air at CC the tube is expanded. The pressure at CC is atmospheric pressure, whilst the pressure at BB is less than atmospheric pressure (as the velocity at BB is greater than at CC). It follows then that the enlargement increases the discharge from the tube, and makes it greater than it would be if the tube were cut off at BB.

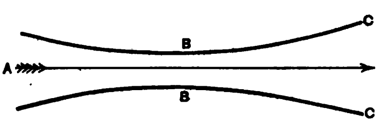


FIG. 37.

Many experiments have been made to determine the effect that is produced by the convergence of the short cylinder to form the jet. The outcome of all these proves that, with a constant head and orifice, the actual discharge begins with a coefficient of ·82 of the theoretical discharge, and increases with the angle of convergence (or contraction of the point of exit from the jet) until it reaches a maximum of ·95, when the angle is $13\frac{1}{2}°$, and sinks to a minimum of ·65. The latter obtains in discharging through a thin plate, when the angle of convergence is at its maximum or 180°.

If, on the other hand, the short cylinder is made to *diverge* at the point of exit, the largest discharge is effected. Venturi's experiments point to the best results being obtained when the

angle of divergence is 5° 6′, with a length of the converging cone nine times the diameter of the smaller base.

If the extremity of the pipe were to have a hollow cone attached to, and around it, and if the combined nozzle and cone were to be surrounded with still water, the outer water would be drawn through the cone, owing to its acquiring part of the velocity of the jet. By directing a jet of water into a fluid, or aeriform medium, currents are induced which have long been utilised in various practical ways, such as in the "Jet Pump" (for drawing off flood waters), in "Giffard's Injector" (for supplying water to boilers), and in the "Steam Injector" or "Blast Pipe."

The application of the hydraulic jet to the propulsion of vessels has been the subject of experiments of a more or less practical form, and of patents dating as far back as 1661. During recent years the system has been tested by the construction of several vessels which were propelled by hydraulic jets, with the result of producing much controversy amongst experts in naval matters. A paper by Mr. Barnaby (read before the Institution of Civil Engineers in 1884) gives a description of the most recent experiment, in the construction for the Admiralty of a torpedo boat by Messrs. Thorneycroft. This was fitted with a turbine propeller, and the design of this boat provided for utilising as much as possible the velocity of the feed water. Just in front of the pumps the bottom of the vessel had a sudden jump upwards from the stern and towards the bow end. At this point the bottom is formed into a great scoop, which gently rises to the inlet of the pump, which is placed at an angle to reduce the effect produced by the change of direction of the feed-water entering. The velocity of this entering water causes it to rise in the scoop, and the vanes of the pump are adjusted to pick up the water without shock, and gradually to accelerate it to the speed of discharge. The peripheral velocity of the pump is 56 feet per second. The energy acquired by the water is utilised by discharging it through nozzles to orifices in the vessel above sea-level. These nozzles

are 9 inches in diameter, formed of copper pipes bent to a radius of 18 inches, and so pivoted that either end can be presented to the discharge orifice in the side of the vessel. The amount of water passed through the pumps in fifteen seconds is equal to the whole displacement of the boat. The water is discharged at a velocity of 37·25 feet per second (about 1 ton per second being discharged with a lift of $21\frac{1}{2}$ feet), the speed of the boat being 21·4 feet per second (or 12·65 knots per hour). Careful experiments were made by means of a thin plate $1\frac{5}{16}$ inch square, attached to the end of a lever and placed in the jet, just where it left the nozzle. The pressure on this plate was recorded by a dynamometer attached to the other end of the lever, and the lever was arranged so as to enable the plate to be shifted about, and the pressure to be recorded over the whole jet. The mean pressure was found to be nine-tenths of that in the centre. Professor Rankine's formula for the efficiency of the jet is as follows:—

$$\text{Efficiency of jet} = \frac{\dfrac{w\,v\,s}{g}}{\dfrac{w\,v\,s}{g} + \dfrac{w\,s^2}{2g} + \dfrac{f\,w\,v^2}{2g}}$$

w = weight of water discharged in lbs. per second;
v = speed of vessel in feet per second;
s = slip or acceleration;
g = 32·2 feet per second.

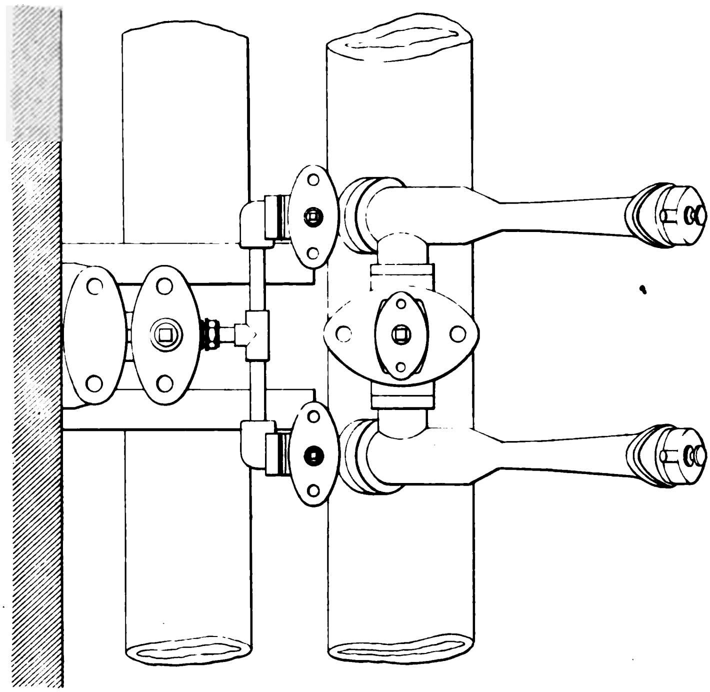
The efficiency of the jet was found to be ·71, and of the pump ·46. The efficiency of the jet and pump combined was ·33, this being the useful work of the jet divided by the effective HP. The total efficiency was ·25, this being the useful work in the jet divided by the indicated HP.

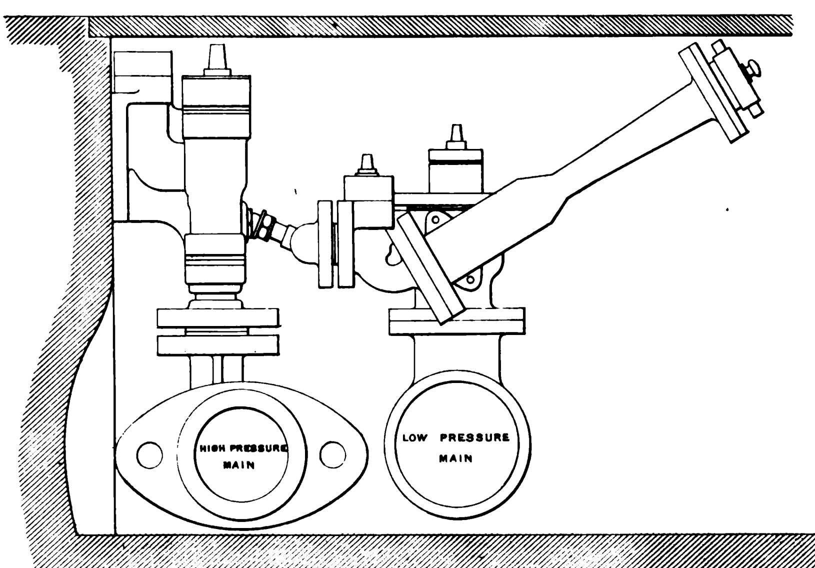
A comparison of the efficiency of a jet propeller with a screw propeller was made by Mr. Barnaby as follows:—

Screw Boat Efficiencies. Engine, ·77; Screw Propeller, ·65; total, ·5.

Hydraulic Boat Efficiencies. Engine, ·77; Jet Propeller, ·71; Pump, ·46; total, ·254.

INJECTOR HYDRANT.





Scale, 1 Inch – 1 Foot.

Inches 12 11 10 9 8 7 6 5 4 3 2 1 0 1 2 Feet

Thos Kell & Son, Lith.

This shows that a jet propeller is a little better than a screw, but that the pump involves a loss of half the power.

Mr. Greathead has perfected an appliance by which a jet of water from a high-pressure main is passed into a stream of water from a low-pressure main, which causes the stream of low-pressure water to be carried to a greatly increased height. If the lines of motion of the two currents were identically the same, so that no loss was sustained from eddies, the force communicated to the mass of water at the low velocity would represent exactly that which the high-pressure water had lost.

Plate 42 shows one of the Greathead "Injector Hydrants" as applied at the Royal Albert Docks, London.

Many experiments have been made to ascertain the quantity of high-pressure water that would be required to produce jets of water at various low pressures. The following table is deduced from experiments with a low-pressure supply from 10 60 to ~~20~~ lbs. per square inch, and with a high-pressure of 700 lbs. per square inch. The quantity given in the table is for a jet of 150 gallons (delivered through a 1-inch nozzle) variously estimated to ascend to a height of from 75 feet to 84 feet, and requiring a pressure of 100 feet head at the back of the nozzle. The length of hose is taken at 200 feet of 2½-inch brigade hose, the resistance of which, for that discharge, is taken at 3 inches of head per foot of hose.

Quantity of High-Pressure or Power Water required for Various Heads of Low-Pressure Supply to produce the Jet described.

Low-Pressure Supply.		High-Pressure at 700 lb. per square inch.
Lb. per square inch.	Feet head.	Gallons per minute.
60	138	3·7
50	115	10·9
40	92	18·1
30	69	25·2
20	46	32·4
10	23	39·6

HYDRAULIC RAM.

The hydraulic ram is a combination of a prime mover and of a pump. The principle on which it is based is, that a body of water flowing from a high to a lower level is capable of exerting an energy sufficient to raise a certain quantity of this water to a level higher than that from which it fell. The machine consists of a chamber into which the supply-pipe delivers water. It has two valves—one a discharge valve opening outwards, and another an automatic pulsating discharge valve, opening inwards and downwards. The latter valve has a weight which is greater than the head of water acting beneath it, so that the valve falls open when the water in the supply-pipe is at rest. This leads to the water flowing through it, and gaining velocity, until it eventually overcomes the weight of the valve, and closes it. The momentum of the water then acts on the discharge valve, which opens outwards, and delivers the column of water to the high level. An air vessel is placed in communication with this outward-opening discharge valve, and the pressure of the water compresses the air in it, by which the shock is relieved and the pressure in the delivery-column is made continuous. By means of a small inward-opening valve (which is attached to a plug close to the outward discharge valve), a little air is sucked in at each pulsation, and passes with the water to the air vessel, thus maintaining the supply of air. If it were not for the interposition of the air cushion, the action of the ram would be attended by a sharp blow each time the current of water reached the necessary velocity to overcome the weight of the valve. A rebound of the water would be produced, which would induce a partial vacuum, with a corresponding opening of the valve, and the admission of a small amount of water. This action would continue, and would produce a number of shocks or vibrations.

In the construction of hydraulic rams the best design pro-

vides for the passage of the water into the machine by easy curves, and not by sharp angles, which produce shocks and consequent loss of energy. There are several hydraulic rams now manufactured which comply with these conditions, and which reach an efficiency of 65 per cent., and even more. About 4 inches is the largest size that the injection-pipe of a ram should be made for safe working.

In the *Transactions of the American Society of Engineers*, Mr. Weston gives the results of some useful experiments which he made to ascertain the effect produced by the sudden closing of valves against water flowing in pipes. Lines of pipes from 1 inch to 6 inches in diameter were laid above ground, and an air vessel was provided which could be connected or disconnected as required. The supply was drawn from a 24-inch main by a 6-inch pipe. The average static pressure in the pipe was 70 lbs. per square inch. In the first series of experiments the water flowed through lengths of pipes of different diameter, thus:—111 feet of 6-inch pipe, 58 feet of 2-inch pipe, and 99 feet of $1\frac{1}{2}$-inch pipe, to a 1-inch outlet pipe, with a $\frac{1}{4}$-inch orifice. In this case the velocity was 0·15 of a foot per second in the 6-inch pipe, and 5·36 feet in the 1-inch pipe. Upon closing the orifice (which was effected in 0·16 of a second) the force of the ram in lbs. per square inch was 129·2 lbs. in the 1-inch pipe, 127 lbs. in the $1\frac{1}{2}$-inch pipe, and 14·5 in the 6-inch pipe. At the dead end of a separate $2\frac{1}{2}$-inch branch-pipe (leading out of the 6-inch pipe at a distance of 300 feet), the force of the ram was 18·8 lbs. With orifices of $\frac{1}{8}$th, $\frac{3}{16}$ths, $\frac{1}{4}$th, and $\frac{5}{16}$ths of an inch, and with velocities of 1·06, 2·57, 5·36, and 6·75 feet per second, the rams in the 1-inch pipe exerted a force respectively of 26·9, 72·8, 129·3, and 158·7 lbs. per square inch. In the 6-inch pipe, with $\frac{1}{4}$-inch and $\frac{1}{2}$-inch orifices, and with velocities of 0·15 feet and 0·53 feet per second, the rams exerted a force of 14·5 and 51·7 lbs.

Mr. Weston made another series of experiments on an

extension of the 6-inch pipe, comprising 182 feet of 6-inch pipe, 66 feet of 4-inch pipe, $3\frac{1}{2}$ feet of $2\frac{1}{2}$-inch pipe, 1 foot of 2-inch pipe, $6\frac{1}{2}$ feet of $1\frac{1}{2}$-inch pipe, and 6 feet of 1-inch pipe. With the $\frac{1}{4}$-inch orifice, and with a flow varying from 0·15 of a foot to 5·39 feet per second in the 6-inch and 1-inch pipes respectively, the ram exerted a force of 4·8 lbs. in the former, and of 66·7 lbs. in the latter. In the 1-inch pipe, with orifices of from $\frac{1}{8}$-inch to $\frac{1}{2}$-inch, the force of the ram increased from 15 lbs. to 177·5 lbs. per square inch. In the $2\frac{1}{2}$-inch pipe, the force of the ram was 22·2 lbs. with a $\frac{1}{4}$-inch orifice, and 183 lbs. with a 1-inch orifice. This latter was reduced to 106 lbs. when the pipes were in connection with the air vessel. In 6-inch pipes the ram (with a $\frac{1}{4}$-inch orifice) exerted a force of 4·8 lbs., and with the 1-inch pipe 80·1 lbs. The latter was reduced to 65·6 lbs. when the air vessel was connected.

PACKING.

For stuffing-boxes for rams, a gasket of hemp, plaited very tight, and well greased, is a very simple and durable packing. After it has become well consolidated the friction is but little, although at first it is considerable. A slight leak serves to lubricate the packing. When the plaiting is done carelessly, the use of hemp is attended with the objection that portions are liable to be torn off when the gland is first packed and worked, and these pieces are liable to get into the valves.

Benjamin Hick introduced the use of cupped leathers into presses, and the experience of his descendant, Mr. John Hick (referred to hereafter), affords valuable data as to the coefficients of friction with leather packing. A cupped leather forms a self-tightening packing, and is very generally used, although it soon wears out and fails when the cup is not properly supported at the bend (where the greatest friction is). This should be done by the insertion of a ring or bush of brass or gun metal, which prevents the rapid wearing away

of the bend of the leather. It has been found that when the cup has been made with a square, instead of with a rounded edge, the joint has not been so water-tight. A form of packing which has been found to answer well in the cylinders of hydraulic capstans is shown in fig. 38. A section of a cup leather packing, such as is used for hydraulic machines, is shown full size in fig 39. FF is the ram working in the cylinder CC. AA is the leather collar which is secured by the wrought iron gland B, recessed into the cylinder, and pressing against a layer of hemp D. To keep the cup open, a ring E, either of brass, plaited hemp, or plaited flax, is introduced. I is a brass guard ring to save the leather at the bend of the cup.

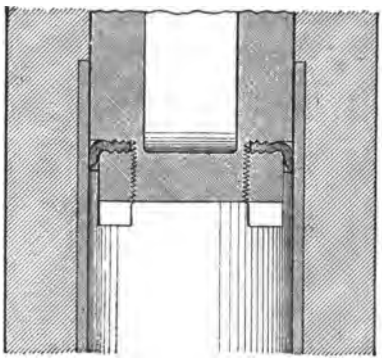


FIG. 38.

Where the cup leather is placed in a shallow (instead of a deep) groove, there is not so much need of the support. The leathers are frequently made far too deep, and this leads to their being more liable to crack, and fail. Gutta-percha or india-rubber cups, and brass or lignum vitae rings, have been used for packing, but on the whole the leather cup is the best.

As the efficiency of hydraulic machines largely depends on proper packing, too much care cannot be taken in seeing that good leather only is used, and that the moulding of the cups is well done. The leather employed for making the cups ought to be of good and close quality, having had oil or tallow well rubbed into it after tanning. Before pressing the leather in moulds, to make the cups, it should be soaked in cold water for twelve hours, and after being forced into the mould it should be left for another twelve hours, then taken out and trimmed, then allowed to dry, and afterwards replaced in the mould for

an hour or two. It can then be removed and dressed to the required shape. The presence of gritty matter in the water injuriously affects leather packing, and involves frequent changing of the cups. Where dirty gritty water has to be used, the leathers wear away very rapidly when the cups are not kept constantly under pressure. If the pressure is taken off hydraulic machines by the accumulator resting on its bed, the water gets between the leather and the ram; and as soon as the accumulator rises off its bed, and the pressure comes on, a little gritty water passes between the leather and the ram, and causes the wear on the packing. An expedient which has been successfully adopted consists in putting a relief valve on the pipe that delivers water to the accumulator from the pumps, and in leaving the suction valve always open. When the relief valve is lifted, at the top of the stroke of the accumulator ram, the pumps being always full of water, the accumulator cannot drop on to its bed. The pressure is in this way constantly kept on the leather, preventing leakage, and at the same time remedying the wear and tear of the packing. The employment of a gun-metal lining to the cylinder has been found to add to the life of a leather packing. The loss occasioned by friction in pumping into an accumulator having a well-packed stuffing-

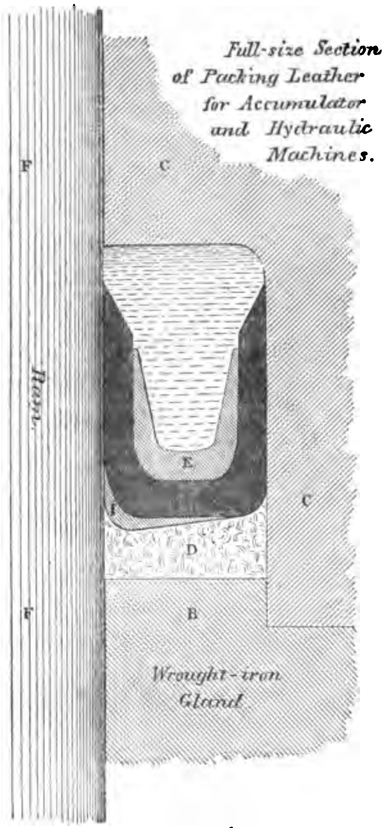


Fig. 39.

box (hemp packing being used) has been found to range from 3 to 8 per cent. at 700 lbs. pressure. The difference of pressure during the rise and fall of the accumulator represents from 1 to 2 per cent. of the power.

Experiments made by Mr. John Hick have afforded data as to friction, from which useful rules have been deduced. He found that friction increases directly with pressure. With leather packing for rams of different diameters, if the pressure per unit of area be the same, friction varies directly as the diameters, or as the square roots of the gross loads. Neither the depth of the leather nor the length of the ram affects the total friction. With hydraulic machines in good order, the amount of friction may be taken to be 1 per cent. for rams of 4 inches diameter, and $1\frac{1}{2}$ per cent. for rams of 8 inches diameter.

From these experiments the following formula is deduced :—

$$F = D \times P \times C$$

Where F = total friction of leather packing.
D = diameter of ram in inches.
P = pressure per square inch.
C = coefficient.

C = ·0471 with new, or badly rubricated, leathers.
C = ·0314 with leathers in good condition, and well lubricated.

The following table of Mr. Hick shows the frictional resistance in percentage of the total hydraulic pressure for rams from 2 inches up to 20 inches in diameter :—

D inches.	F per cent.	D inches.	F per cent.
2	2·00	12	0·33
3	1·33	13	0·30
4	1·00	14	0·28
5	0·80	15	0·26
6	0·66	16	0·25
7	0·57	17	0·23
8	0·50	18	0·22
9	0·44	19	0·21
10	0.40	20	0·20
11	0·38		

$$F = \frac{0{\cdot}0416}{D} P \left\{ 0{\cdot}09347 - 0{\cdot}00000019291 \, P \right\}$$

Where F = friction in lbs.
D = diameter of piston in feet.
P = gross pressure in lbs. per square foot of piston area (correct up to about 1 ton per square inch).

POWER CO-OPERATION.

The concentration at one or more points of the power necessary for the supply of water for domestic purposes (in the same way as gas and other requirements of daily life are produced from central stations) has suggested the desirability of developing at one establishment the power that is requisite to actuate machines at work in an area within reach of such centre. The principle has been termed by the author "Power Co-operation," and after advocating it for many years it is now gaining ground steadily. The facility with which power can now be transmitted to great distances, enables the co-operation of many power-consumers to be brought about. Bramah realised its feasibility in the following characteristic letter, which was written by him to Robert Mallet in 1802. Referring to the hydraulic press, Bramah wrote—"I think much might be done in Ireland in the press way if the excellence of the principle was but known; I have also now applied it with the most surprising effect to every sort of crane for raising and lowering goods in and out of warehouses. So complete is the device, that I will engage to erect a steam-engine in any part of Dublin and from it convey motion and power to all the cranes on the quays and elsewhere, by which goods of any weight may be raised at one-third of the usual cost. This I do by the simple communication of a pipe, just the same as I should do to supply each premises with water. I have a crane on my own premises which astonishes every person to whom it has been shown—as they see the goods ascend and descend fifteen or twenty times in a minute to the height of 18 or 20 feet, and at the same time it is impossible for any person unacquainted with the principle to discover how or where the power comes from. I also show them pumps raising water with huge force, and a press squeezing wood, &c., to atoms, and not the

smallest discovery can be made of the cause. I believe I shall have all the cranage of the London wet-dock warehouses to undertake, which will be the grandest job perhaps ever done."

More than twenty years ago the author promoted an Hydraulic Power Co-operative undertaking in Manchester, and the late Sir W. Fairbairn wrote in regard to it as follows:—

"Your proposal to erect steam-engines and lay down pipes for the purpose of working accumulators for supplying hydraulic power to different localities of the city of Manchester, seems to have several advantages over the system now in use in the different warehouses where steam is employed. In the first place, it would remove steam-engines and boilers from the premises, lessen the risk from fire and boiler explosions; and secondly, it would supply the necessary power to work cranes, hoists, hydraulic presses, &c., in those depôts on principles of increased security."

The experience of many years of the successful working of hydraulic machinery and appliances in government dockyards, railway termini, docks, wharves, &c., has proved it to be a most economical, convenient, and safe method of transmitting power over great distances, where the requirements are intermittent, as in the case of loading and discharging ships and railway trucks, lifting and lowering goods at wharves and warehouses, working dock-gates, packing or pressing goods, actuating tools and machines in works. For these and for many other operations the number of labourers can greatly be reduced, as a system of power co-operation admits of the concentration of the whole engine and boiler power that are requisite to supply hydraulic pressure to an entire district at one or two points, by which the power is accumulated, and is distributed, at the minimum cost and trouble. Independent engines, boilers, and attendants can be dispensed with. The buildings and spaces occupied by these engines and boilers are rendered available for other purposes. The danger

of fire due to the presence of boilers is avoided. The power is always available to meet the requirements of the consumer, and at a cost to him only in proportion to the power absolutely consumed.

Hull was the first town in which high-pressure hydraulic mains were laid under the public streets for the supply of water-power on the co-operative system, and there the author carried out the first hydraulic power installation. The following description of these works was given in a paper read at the Institution of Civil Engineers in 1877 ("Robinson on the Transmission of Power to Distances"):—

"In the year 1872 an Act of Parliament was obtained for the purpose of establishing, at Kingston-upon-Hull, what was termed in the preamble 'a system for applying motive power by hydraulic pressure to waterside and land cranes, used for the purpose of raising and landing goods; and for working dock gates and other machinery.' The powers granted under this Act were to be exercised over an area of 60 acres, and they authorised the abstraction from the old harbour of the river Hull (a tributary of the river Humber) an amount of water not exceeding 1,000,000 gallons a day, for distribution within the company's district, for which a payment was to be made to the Corporation of £12, 10s. per annum for each 250,000 gallons; the water to be used for no other purpose than as a motive power, except with the consent of the Corporation.

"A 6-inch pressure main has been laid from the northern boundary of the defined area, near the Cottingham Drain, in a southerly direction along Wincolmlee, Trippett, Dock Office Row, under the Old Dock Basin (which forms the eastern or river Hull entrance to the Queen's Dock), and crossing this entrance it is laid along the whole length of High Street, terminating close to the western approach of the South Bridge across the river Hull. The length of pressure main, exclusive of the dock crossing, is altogether 1485 yards, that on the

north side of the dock entrance being 673 yards in length, and that on the south side 812 yards. Except at the dock crossing the main consists of cast iron flanged pipes, of 6 inches internal diameter, 1 inch thick, with the usual spigot and faucet, and with gutta-percha ring joints tested to 2800 lbs. per square inch before being laid, and afterwards to 800 lbs. per square inch.

"Stop-valves at intervals, having a waterway equal to that of the main, divide the main into sections. Air-cocks are fixed on all summits, by which the air is displaced in charging the main. T-pieces for 2-inch, 3-inch, and 4-inch branches are placed at convenient points, from which service-pipes can be carried to the various warehouses, works, &c.

"The main was laid across the dock entrance, in a trench dredged to the inverts forming the dock bottom, the solid obstructions met with being removed and the bottom levelled by a diver. The pipes across the dock are of 6 inches internal diameter, made of welded wrought iron $\frac{3}{4}$-inch thick, bent to template to suit the curves of the sides and bottom of the dock, and were tested to 3000 lbs. per square inch at the manufacturer's. They were put together at the side of the dock, and tested to $\frac{1}{2}$ ton to the square inch before being lowered into the trench. This was done from barges, and when the pipes were in position they were well concreted, to protect them from being injured by anchors or by weights falling overboard from ships. This part of the work has been tested in an unexpected way by the stranding of a large ship over the pipes, which, however, were in no way injured.

"The power to supply the water-pressure is concentrated at one pumping-station in Machell Street, where an engine-house has been built to receive four 60 HP engines. The ground being silty and bad, the foundations were carried down to the hard clay, a depth of 24 feet, the walls being built on arches resting on concrete piers. The engine-house is covered by a tank fitted with filtering boxes, through which the water

pumped from the river Hull passes before it is delivered to the engines. Two pairs of high-pressure horizontal pumping-engines have been erected, each engine being of 60 HP, and capable of pumping 130 gallons per minute at 700 lbs. pressure per square inch, with steam at 100 lbs. pressure. The steam cylinders are $12\frac{1}{4}$ inches in diameter, and the length of stroke 24 inches; the force pumps, which are double-acting, have a $4\frac{9}{16}$-inch piston, the piston rod being $3\frac{1}{8}$ inches in diameter. Space is provided in the engine-house for two additional pairs of 60 HP engines, which can be erected at a future time when the demand for the water-pressure requires further engine power. Two Lancashire boilers, 22 feet 6 inches long, and 6 feet 6 inches in diameter, supply steam to the engines.

"An Appold centrifugal pump (in duplicate), fixed in the engine-house, draws the water from the river Hull, a distance of 125 yards, through a 10-inch pipe, and delivers it into the tank, the lift being 35 feet from low tide. The pump has an 8-inch suction, and is driven by a Brotherhood's 4-inch three-cylinder engine, also in duplicate. Each engine and pump supply 800 gallons of water per minute, with 100 lbs. steam pressure. A 6-inch return pipe is laid from the tank to the river, serving both as an overflow pipe and as a means of cleaning out the tank.

"One accumulator is erected at the pumping station in Machell Street. It has a diameter of 18 inches, and a stroke of 20 feet. The case is loaded with $57\frac{1}{2}$ tons of copper slag and sand, which produce a pressure of 610 lbs. per square inch in the main. Provision is made for an additional accumulator at the pumping station when required. Another accumulator will be placed at Grimsby Lane, towards the southern extremity of the line of main.

"Several observations were made to ascertain the useful effect of the engines and accumulator, and the mean was found to be 76 per cent., 5 per cent. being the loss in the pumps.

"Each crane will have a counter attached to it to register the amount of work done. One hundred tons may be lifted 40 feet, or 200 tons 20 feet, and so on each day by each crane for the above charge, which is under ½d. per ton for a lift of 40 feet. If more work than this is done, the extra work will be charged at the rate of 4s. for every additional 100 tons lifted 40 feet. Special rates will be made for working presses, hydraulic engines, capstans, small cranes, &c., as occasion arises."

These works at Hull have been continuously and successfully in operation since they were completed, and have established the practicability of the system, which has been followed in the metropolis, Messrs. Ellington & Woodall being the engineers.

The extent to which the transmission of fluid through pipes has been carried out in America deserves special mention when speaking of power co-operation. In order to enable petroleum to be brought to the various commercial centres, independently of the ordinary channels of transport by road, rail, or river, a system of transmission through pipes was commenced in 1865. Oil was pumped from Pithole, in Pennsylvania, towards Oil Creek, a distance of 3200 feet. Later in that year, another line of pipes was laid, through which oil was pumped a distance of seven miles. The system grew, until in 1875 a 4-inch pipe was laid from the lower oil country to Pittsburgh, Pennsylvania, a distance of about sixty miles. This main (as in previous cases) had to be guarded by armed men, to protect it from injury by those interested in other methods of transport. The system became ultimately recognised as the best method of conveying the oil from the place of production to the place where it was to be distributed; and the various lines of pipes became amalgamated (with a single exception) in one company, known as The National Transit Company. The length of pipes under the control of this company can be judged by the following

figures :—The New York pipe is 300 miles; the Philadelphia pipe is 280 miles; the Baltimore pipe is 70 miles; the Cleveland pipe is 100 miles; the Buffalo pipe is 70 miles; the Pittsburgh pipe is 60 miles. The sizes of the pipes vary from four to six inches. The entire length of these pipes (including duplicate lines) is upwards of 1300 miles; while the length of the collecting pipes of two inches diameter in the oil region is estimated to be from 8000 to 10,000 miles.

COST OF HYDRAULIC POWER.

The following data with reference to the cost of hydraulic power were given in the before-mentioned paper by the author on the "Transmission of Power to Distances":—

"In the Albert Dock of the Hull Dock Company, a 60-HP engine supplies water for working an 80-feet swing-bridge, nineteen hydraulic engines working gates, sluices, and capstans, three 20-ton coal hoists, one 15-ton crane, one 3-ton crane, and thirty-four $1\frac{1}{2}$ ton cranes. These machines are worked at a pressure of 775 lbs. per square inch, through 5350 feet of 5-inch pipe, 1400 feet of 4-inch pipe, with 3-inch and 2-inch branches to the dock gates and warehouses. The cost of supplying water power for the year 1875 was £1367 3s. 1d., which gives, after taking 80 per cent. as the useful effect of the water after delivery into the main :—

	d.		
Engine power . . .	0·24	per 100	foot-tons.
15 per cent. for interest on capital and depreciation	0·88	,,	,,
	1·12	,,	,,
Add for repairs .	0·13	,,	,,
	1·25	,,	,,

"At Cotton's Wharf, London, there are ten 25-cwt. hydraulic

cranes lifting 40 feet, four 2-ton single power cranes, one 4·2-ton double power crane and one 48-ton press worked at a pressure of 700 lbs. per square inch; the cost when only six cranes were in operation, which is the average number, was:—

	d.	
Engine power . . .	0·63	per 100 foot-tons.
15 per cent. for interest and depreciation . .	1·26	" "
	1·89	" "

"The cost of labour at the cranes was 0·46d. per 100 foot-tons. If the whole of the sixteen appliances were working, the cost would be:—

	d.	
Engine power	0·23	per 100 foot-tons.
15 per cent. for interest and depreciation . . .	0·47	" "
	0·70	" "

"The labour at the cranes being the same as before, namely, 0·46d. per 100 foot-tons.

"At the St. Katherine Docks, engines of 140 HP, nominal, pump 5,000,000 cubic feet of water annually at 600 lbs. pressure through 1200 yards of 7-inch main, supplying power to work a swing-bridge and upwards of seventy-five cranes, hoists, and presses. The power exerted annually is nearly 193,000,000 foot-tons, or, taking 80 per cent. efficiency, more than 154,000,000 foot-tons.

	£.
The cost of the engines, boilers, accumulators, pipes, and appliances	= 35,000
Foundations of engines and boiler-house . .	= 12,000

"The cost of water delivered into the main, including coal, wages, repairs and supervision, is 10s. per 1000 cubic feet. The cost of the water power is therefore as follows:—

	d.	
Engine power	= 0·39	per 100 foot-tons.
15 per cent. for interest on capital and depreciation	= 1·10	" "
	1·49	" "

"At the London Docks, engines of 185 nominal HP pump 7,000,000 cubic feet of water per annum, of which 4,250,000 cubic feet are pumped at 750 lbs. pressure through 1450 yards of 5-inch pipe, 640 yards of 4-inch, and terminating with 550 yards of 3-inch. The remaining 2,750,000 cubic feet are pumped at 650 lbs. pressure through 750 yards of 6-inch pipe. These jointly work the swing-bridges, lock gates, and upwards of eighty cranes, hoists, presses, &c.

"The cost of water delivered into the main, including coals, wages, repairs, and supervision, is 10s. per 1000 cubic feet. The cost of the power will therefore be as follows:—

	d.	
Engine power . . .	0·33	per 100 foot-tons.
15 per cent. for interest and depreciation . . .	0·88	" "
	1·21	" "

"At the Victoria Docks, engines of 280 nominal HP pump 8,000,000 cubic feet per annum at 780 lbs. pressure through 700 yards of 5-inch pipe, 2000 yards of 4-inch, terminating with 200 yards of 3-inch pipe. The power exerted is 401,000,000 foot-tons, or 321,000,000 foot-tons at 80 per cent. efficiency; and this power is applied to working a swing-bridge, lock gates, capstans, and upwards of one hundred cranes and hoists.

"The cost of water delivered into the main, including coals, wages, repairs, and supervision, is 10s. per 1000 cubic feet. The cost of the power will therefore be as follows:—

	d.	
Engine power . . .	0·30	per 100 foot-tons.
15 per cent. for interest and depreciation . . .	0·88	" "
	1·18	" "

"At the Great Western railway station at Paddington, a 70-HP engine supplies water at 700 lbs. per square inch to two

waggon hoists, three hauling machines, twenty turntables, fifty-four 25-cwt. cranes, sixteen hoists, three capstan engines, three traversing tables, two drawbridges, one ticket-printing machine, and four dropping platforms. According to Mr. H. Kirtley, the average consumption of water is 25,600,000 gallons per annum, obtained from the Water Company at 4d. per 1000 gallons, one-fourth being returned and three-fourths run to waste. The cost of supplying this appears to be 1·10d. per 100 foot-tons, taking 80 per cent. efficiency of water delivered, and allowing 15 per cent. for interest and depreciation, or adding 0·13d. per 100 foot-tons for repairs = 1·23d. per 100 foot-tons.

"At the Swansea Docks, the amount of water pumped in the year ending midsummer 1876 was 20,750,000 gallons, at 700 lbs. per square inch, and the working expenses were:—

	£	s.	d.
Coal and fuel	1056	19	9
Stores	134	15	5
Wages	699	14	1
	1891	9	3
Wages and repairs	412	6	10
Materials	244	7	6
	656	14	4

"The cost will therefore be, taking 80 per cent. for the useful effect of the water delivered into the main:—

	d.	
Engine power	0·38	per 100 foot-tons.
15 per cent. (on £22,000 for interest and depreciation .	0·66	" "
	1·04	" "

"The extra cost for wages, repairs, and materials would be 0·13d. per 100 foot-tons, making the total cost 1·17d. per 100 foot-tons.

"The following is a summary of the foregoing data, and represents the cost of water power at pressures varying from 600 to 780 lbs. per square inch, taking 80 per cent. as the

efficiency of the water pressure after delivery into the main, and allowing 15 per cent. for interest and depreciation.

	d.	
Albert Docks, Hull	1·25	per 100 foot-tons.
Cotton's Wharf (maximum)	1·89	,, ,,
Cotton's Wharf (minimum)	0·70	,, ,,
Paddington	1·23	,, ,,
Swansea	1·17	,, ,,
St. Katherine Docks	1·49	,, ,,
London Docks	1·21	,, ,,
Victoria Docks	1·18	,, ,,
Mean	1·26	,, ,,

The following tariff (issued by the Hull Company at the outset), gives the rates at which it was proposed to supply the water power:—

	£	
1 crane in one warehouse	52	per annum.
2 cranes ,, ,,	94	,,
3 ,, ,, ,,	132	,,
4 ,, ,, ,,	166	,,

The rates charged by the Hull Company in 1885 were as follows:—

Nearly all pay 4s. per 1000 gallons with a minimum of £5 per quarter, whether 25,000 gallons have been used or not. Some pay 8s. per 1000 gallons with a minimum of £2, 10s. per quarter whether 6250 gallons have been used or not.

In the metropolis the charges made by the Hydraulic Power Company are based on a minimum payment of 25s. per quarter for each machine, and a sliding scale for the water, which is measured by meter. The following is the scale of prices:—

Under 3,000 gallons per quarter, 25s. per machine.

	Gals.		Gals.			Gals.
Above	3,000,	and not exceeding	5,000	per quarter,	8s.	per 1000
,,	5,000	,,	10,000	,,	7s.	,,
,,	10,000	,,	20,000	,,	6s.	,,
,,	20,000	,,	50,000	,,	5s.	,,
,,	50,000	,,	100,000	,,	4s.	,,

,, 100,000 special terms.

The total cost of producing and distributing high-pressure water power at 700 lbs. pressure may be taken at from 6s. 6d. to 8s. per 1000 cubic feet, according to the price of coal, and other varying circumstances.

Mr. B. Walker has estimated that the cost of producing 1000 cubic feet of water at a pressure of 700 lbs. per square inch is 6s. 7d. by the ordinary high-pressure non-condensing and non-expansive pumping-engine. With economical highly expansive surface-condensing engines the cost is reduced to about 4s. 3d. These figures are based on the coal costing 8s. per ton, and they include attendants' wages, interest, and 5 per cent. on the capital expended upon engine-house, foundations, engines, boilers, accumulator, and pipes, together with the cost of the water for the boilers.

Reducing the above rates to the cost of 100 foot-tons, and allowing 80 per cent. as the useful effect of the water power, the price of 6s. 7d. per 1000 cubic feet is equivalent to 0·22d. per 100 foot-tons, and that of 4s. 3d. to 0·142d. per 100 foot-tons. Under some circumstances the cost is higher, and the price would be 8s. in the first case, and 6s. 6d. in the second. 8s. per 1000 cubic feet, reduced in the same way as before for comparison, is equivalent to ·267d. per 100 foot-tons, and 6s. 6d. is equivalent to 0·22d.

Mr. Westmacott calculated that the cost of producing 1000 cubic feet of water at 700 lbs. pressure at the Poplar Docks, London, was 6s. 7d. in the year 1878, was 5s. 7d. in 1879, and was 4s. 11½d. for the first half of 1880. The number of gallons pumped was at the rate of 55¼ million per annum in the first case, 69½ million in the second, and 77¾ million in the third. The work was done by six ordinary high-pressure engines, coal costing about 14s. 6d. per ton. Reducing these prices in a similar manner to those previously given, the cost per 100 foot-tons becomes respectively 0·22d., 0·186d., and 0·165d. To these figures must be added about 1s. 6d. per 1000 gallons (equivalent to 0·313d. per 100 foot-tons) for

wear and tear, and interest on capital, in 1878, and a proportionately smaller amount for the years 1879 and 1880. As the amount of work done increased, the cost per 100 foot-tons diminished, and since at the time to which the statement referred the engines were not worked up to their full power, the cost would be probably still further reduced.

At Cardiff Docks with ordinary engines worked up to their full power, the cost (including fuel stores and working expenses) was 3s. per 1000 cubic feet, equivalent to 0·1d. per 100 foot-tons.

TAPPING PRESSURE MAINS.

An ingenious plan for tapping low pressure mains without cutting off the water has been devised by Mr. Morris. This is shown by fig. 41. X is the pipe to be tapped, and to have a junction ferrule inserted, under pressure. H is a packing piece, which is first placed on the pipe over the point where the hole is to be made, a layer of greased felt or other material being laid between the two, to make a good joint. A saddle piece D is then fixed by means of a chain F. A spindle A, with a drill tap B at its lower end, is inserted in the top of the saddle piece. A socket E, having a leather washer at its lower end (to prevent leakage), has, resting in the top of it, the nut C, with arms for screwing in, and for unscrewing the saddle screw. After the hole is tapped in the pipe the drill tap is drawn up, and the solid part of the slide G (through which the tap passed) is shifted over the hole thus tapped in the packing piece H. By this means the water cannot escape when the spindle A and the drill tap B are removed. The drill tap is then taken from the socket of the spindle A and is replaced by a screw plug M, which projects below the socket, and to the lower part of this is screwed the ferrule I, containing the plug valve. The spindle, ferrule, and plug are

then introduced into the saddle-piece, and the nut C is screwed up again. After this the slide G is drawn from over the hole, and the lower part of the ferrule is screwed into the hole that has been drilled, and tapped, to receive it. When this is done the whole is removed, leaving only the lower part of the ferrule (containing the plug valve) in the pipe. The upper part of the ferrule (having the top caps screwed

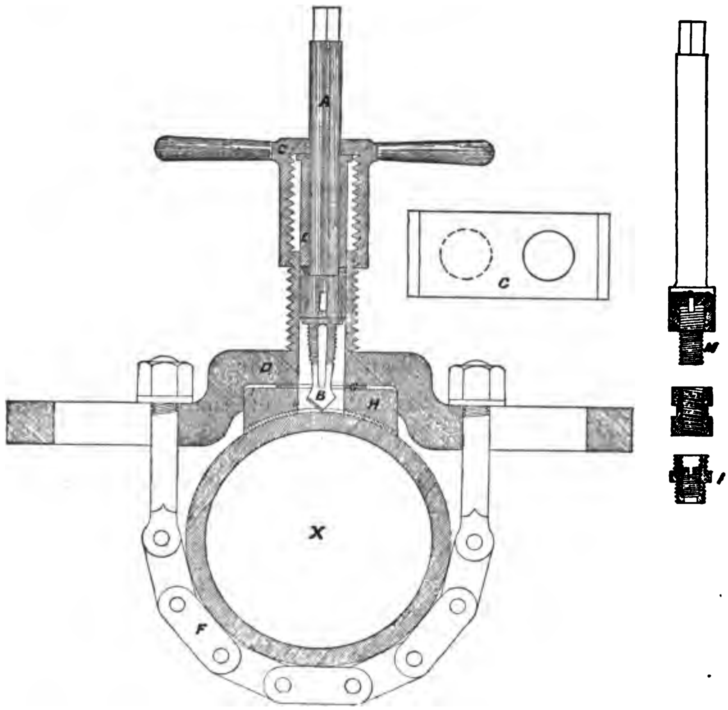


FIG. 40.

down) is then screwed on to the lower part, and the service pipe is connected to the outlet. The internal plug valve is then unscrewed by a small tee spanner inserted through a hole in the top cap. As the plug is unscrewed from the lower part of the ferrule, its upper end enters and screws into the upper part, thus closing it, and opening a communication between the inlet and outlet ferrule.

METERS.

Many ingenious meters have been invented to enable the quantity of water flowing in a pipe to be automatically measured. Some of these (such as Parkinson's) are based on the principle that is employed for gas meters, the water being admitted to one side of a drum, which is divided into segmental compartments. This drum is caused to rotate by the entry of the water, which passes through it to the delivery. The quantity of water that each compartment holds being known, the volume that is passed by each revolution is measurable by a train of wheels connecting with a clock face.

The Pocock Meter is made with three single-acting diaphragms which are impervious to water. These actuate a three-way single slide valve. The water passes into the measuring spaces between the diaphragms and the cover of the meter, and presses the diaphragms towards the centre, one after the other, causing the rotation of the crank, and the registration of the volume of water.

Another class consists of piston meters. In these the water, passing through a cylinder, forces up and down a piston, which actuates an index and records the volume that is passed. The admission and outlet are either through a four-way cock or through slide valves. Two types of this form of meter are Kennedy's and Frost's. In the Kennedy Meter the stroke of the piston is reversed by a tumbler turning a four-way cock, through which both the supply and delivery water passes. In the Frost Meter the reciprocating action is arranged by means of slide valves and ports reversed by a tappet on the piston rod. The piston in the Kennedy Meter is packed with a rolling ring of indiarubber, and in the Frost Meter the piston is packed by double cup leathers.

Another piston meter is that known as Barton and West's. In this, the water enters the cylinder through valves which are

reversed by the stroke of the piston. When the water is acting on the top of the piston, it forces the water out from the bottom, so that it rises up in a space in the case outside the cylinder in which the piston works, and passes away above through a valve chamber. At the end of the stroke the valves are reversed, and the piston rises by the pressure of the water, which reaches the under side by flowing down the before-mentioned space surrounding the piston cylinder.

An application of the Reaction Turbine, or Barker's Vertical Mill, to the purpose of measuring water was made by the late Sir W. Siemens, and is called a Turbine Meter. In this appliance the water is admitted (by a fixed inlet) through a contracted entrance pipe to the top of the horizontal rotating wheel, and passes from it through spiral channels inside the drum, the reaction of the water producing a rotating action. This is regulated by vanes projecting from the revolving drum, which are adjusted so as to insure that the speed of rotation corresponds to the rate of flow, and is not disturbed by the velocity.

Another form devised by Sir W. Siemens is the Fan Meter. In this the current of water acts on the circumference of the blades of a fan, which is caused to rotate by the water passing upwards through oblique openings on to the blades. The water afterwards flows away through a pipe in the top of the case above the fan. In the case are placed projecting plates to regulate the flow, by adjusting their sizes in proportion to the velocity.

Another application of the fan system is that used in the Tylor Meter. The water is directed upon the blades of the fan upwards through oblique vertical openings beneath, and after it has acted upon the fan it escapes at the same level. A feature in this meter is, that two or more blades are in the direct route of the water passing to and from the meter. It could be used as an automatic recording meter, as, by placing it on a water-main close to a valve, the leakage of water would

be detected at nights, when the district supplied by the main should be drawing no water.

Another piston meter is the Galasse, which is made by a Belgian manufacturer. It is constructed on the principle of a pumping-engine, having two double-acting water cylinders with pistons. The water is admitted through ordinary three-ported side valves, which are actuated by the piston rods of the alternate cylinders. The pistons and glands are packed with leather rings. At the Health Exhibition of 1884, there were several other meters shown, made by Muller, Meinecke, Paine, Keystone.

There is a risk in some meters of a certain quantity of water passing without being registered, whether in the rotary meters, or in the drum and piston meters. The risk of this has deterred many from applying meters to the registration of water when supplied under pressure, either for motive power, or for domestic supply.

A means of detecting a leak in a water-main, arising either from a defect in the pipe itself, or from a consumer's service-pipe not being properly shut off, has been devised by Mr. Deacon. The appliance, which is termed a "Waste Water Meter," can be placed either directly on the main, where it is easily accessible, or upon a branch pipe between two points on the main. The branch pipe can be carried under the adjoining footway, where it would not be easy to examine it under the roadway. The feature of the meter is that it differentiates the flow, and exhibits this upon a diagram, traced upon a recording drum, which is caused to revolve once in twenty-four hours by clock-work, placed in a dry chamber above the body of the meter through which the water passes. A pencil rises and falls in contact with the drum (according as water is passing or not), and by means of the rotating drum a diagram is obtained which affords a perfect record of both the rate of flow, and of the time of day or night when it occurs. If water is steadily running out of the main, through

a leak, or through taps being left open, it is indicated by a horizontal line traced on the paper placed on the drum. Vertical ordinates of the diagram indicate rises and falls in the rate of flow. By attaching this meter to the mains, in a district where water is suspected to be wasted, the particular locality can be detected, and the leakage investigated. The employment of this meter has resulted in the detection of the cause of waste of water in Liverpool (where it was introduced), and it affords a means of facilitating the labours of those who are responsible for the economical distribution of water, either for domestic or for manufacturing purposes.

In order to equalise the pressure of water in the mains throughout a district having great variations of level, an appliance has been devised which is called "Key's Pressure Reducing Valve." This valve is of a globular form, and has a diaphragm cast in it, cutting off the high-pressure supply water from that in the low-pressure delivery pipe. This diaphragm is of irregular shape, its general direction being diagonally upwards from the bottom of the supply towards the top of the delivery pipe. It has a horizontal circular aperture, upon the circumference of which stands a fixed cylinder with holes in its sides, open at the bottom, and reaching to the cover of the valve. Inside this fixed cylinder another cylinder, with both ends open, is placed free to move up and down in it. To the top of this moving cylinder is fixed, by means of a crossbar, a piston-rod, which passes upwards through another cylinder of smaller diameter. To this rod is attached a piston which works watertight in the smaller cylinder, the rod itself, which passes in through the top of the valve, being loaded with weights. The action of the valve is as follows:—If the pipe is empty, the weights press the piston, &c., down upon a seating, by which through the supply pipe water is admitted. This encounters the diaphragm, and, passing upwards, surrounds the first-mentioned cylinder. It flows through the holes in the sides of this, goes through the moving

cylinder into the lower part of the globular valve, and so out into the delivery pipe. When the latter is full, and the water begins to exert pressure, it presses upon the lower face of the piston, giving it a tendency to rise. When the pressure reaches a point at which it can overcome the weights that hold down the piston, it forces up, and the rising piston carries with it the moving hollow cylinder which then (as it rises) closes the holes in the fixed cylinder, and thereby shuts off the water in the supply from that in the delivery pipe. By this arrangement the pressure can be reduced to any required extent by varying the weights upon the valves.

Another form of valve to effect the same purpose is Barton & West's Water-Pressure Reducer. In this valve, water from a supply pipe passes into a chamber in which is a vertical piston rod with a piston head at top and bottom. The water acting on the under surface of the top piston, and on the upper surface of the bottom piston, balances the two, so that the flow of water has no tendency to raise or lower the rod, which passes upwards through the top of the valve chamber, and is acted upon by a weighted lever which tends to press it downwards. In so doing, it opens a circular aperture against which the lower piston closes when the valve is shut. When the valve is opened it allows water to pass through the aperture into a lower chamber, from which it flows into an outlet pipe. In the lower chamber the water presses against the lower surface of the bottom piston (or rather upon a portion of its surface, which is reduced in area by a cylindrical prolongation of its central portion) tending to force it up against the aperture. As long as this water-pressure in the lower chamber is less than that which acts upon the piston from the weighted lever, the piston will be held down. Water will then pass from the supply pipe through this aperture into the lower chamber, and thence into the outlet pipe. As soon, however, as the pressure in the lower chamber exceeds that transmitted by this lever, the piston will be forced upwards, and will close

the aperture. By regulating the weighting of the lever the pressure can be reduced to any required amount. The valve chambers are sealed at top and bottom by discs of india-rubber, through which the piston rod passes, and which render any packing of the pistons unnecessary, as these india-rubber discs prevent leakage.

A "Water-Pressure Regulator" has been devised by Mr. Foulis. It is divided into an upper and lower chamber by means of a diaphragm, which passes from the upper side of the inlet in a diagonal direction to the under side of the outlet. In the centre is a circular aperture, the upper edge of which forms the valve seating. Vertically, under this aperture, and at the bottom of the chamber, is a short cylinder from which a small

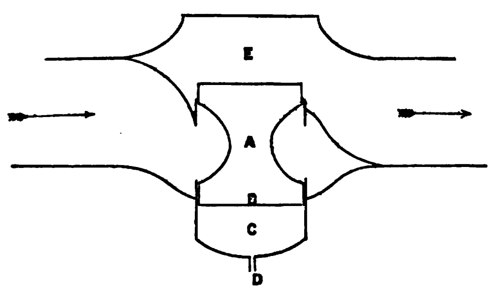


FIG. 41.

pipe communicates with the regulating apparatus, which is shown by the sectional diagram, fig. 41. The valve A (formed to give a gradual opening) has the lower portion B acting as a piston working in the cylinder C. The pressure from the inlet, acting both underneath the valve and above the piston, is neutralised, and has no effect in moving the valve. A small pipe D communicates with a regulating apparatus, and gives the required pressure upon the lower side of the piston. Whatever pressure is applied below the piston, an equal pressure will be maintained above the valve, and in the outlet pipe. The pipe D is carried to a small hydraulic press having a ram about $\frac{1}{2}$ inch in diameter, loaded with weights equi-

valent to the pressure that is desired in the chamber E. Another pipe puts this press in communication with the main inlet pipe, and it is also connected with a waste-pipe. The press and the various pipes are connected, so that as long as the pressure in C does not exceed that desired in E, the ram remains stationary at its lowest point; but when this pressure is exceeded, the ram rises, thereby opening a communication between the pipe D and the waste-pipe, which immediately relieves the pressure in C, whereupon the valve A falls. This process is repeated as often as the pressure in C (and consequently in E) rises above that to which the hydraulic ram is set. Any rise in the outlet pressure, due to lessened consumption, will of itself close the valve A.

A modification of this valve is to make the piston B of smaller diameter than the valve, and to leave the chamber C open to the atmosphere. The separate regulating apparatus can then be dispensed with. In this case the inlet pressure on the valve is not completely balanced; but, acting on the greater area of the valve, tends to lift it, and will do so until the total outlet pressure on the top of the valve is equal to the total unbalanced inlet pressure.

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